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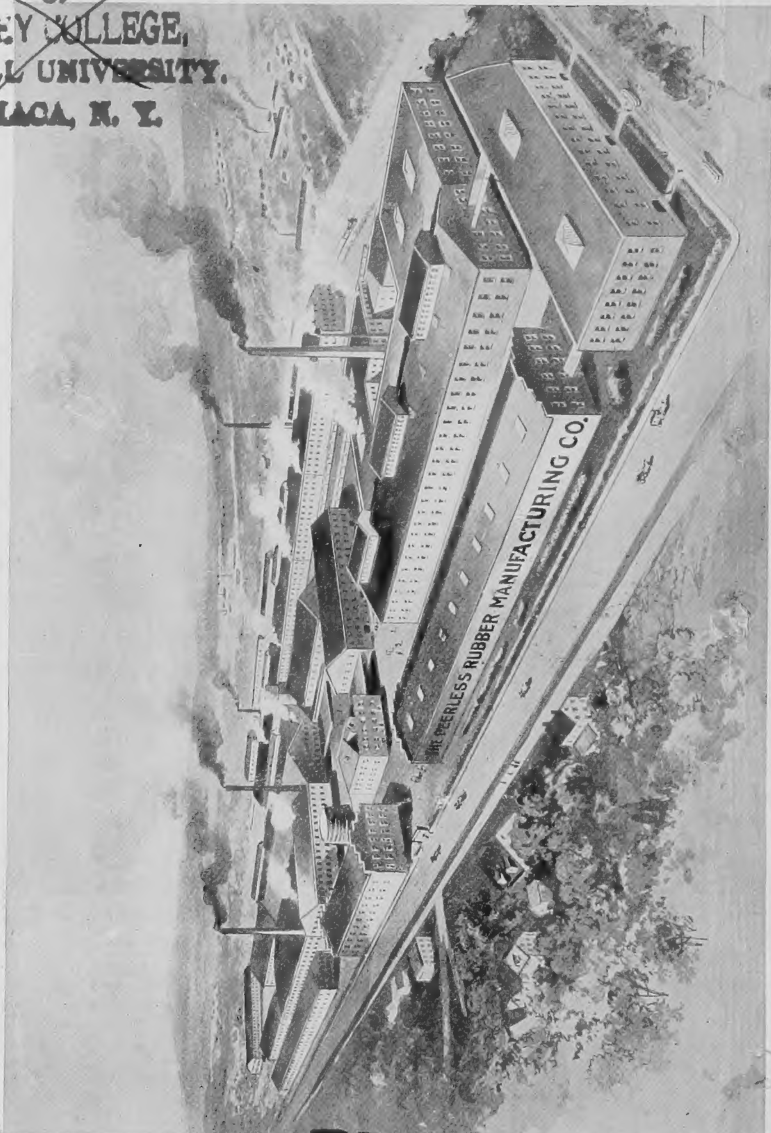

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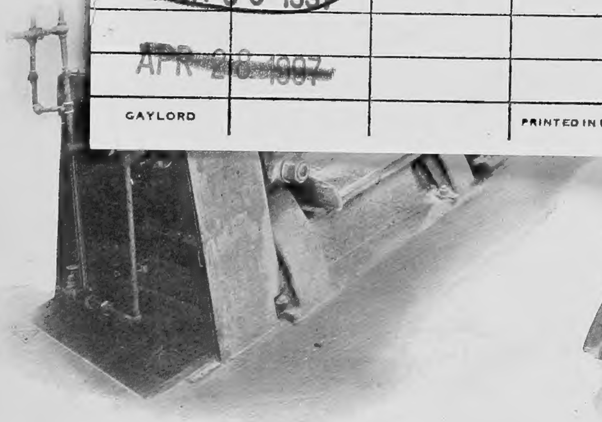

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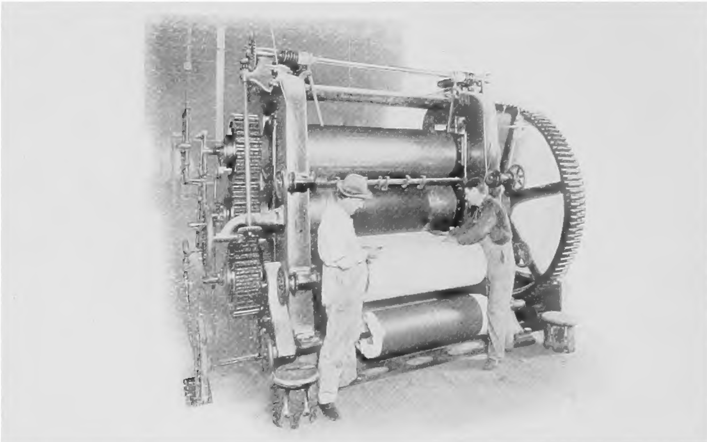


SHEETING OUT COMPOUNDED RUBBER

MANUFACTURING HOSE BY HAND

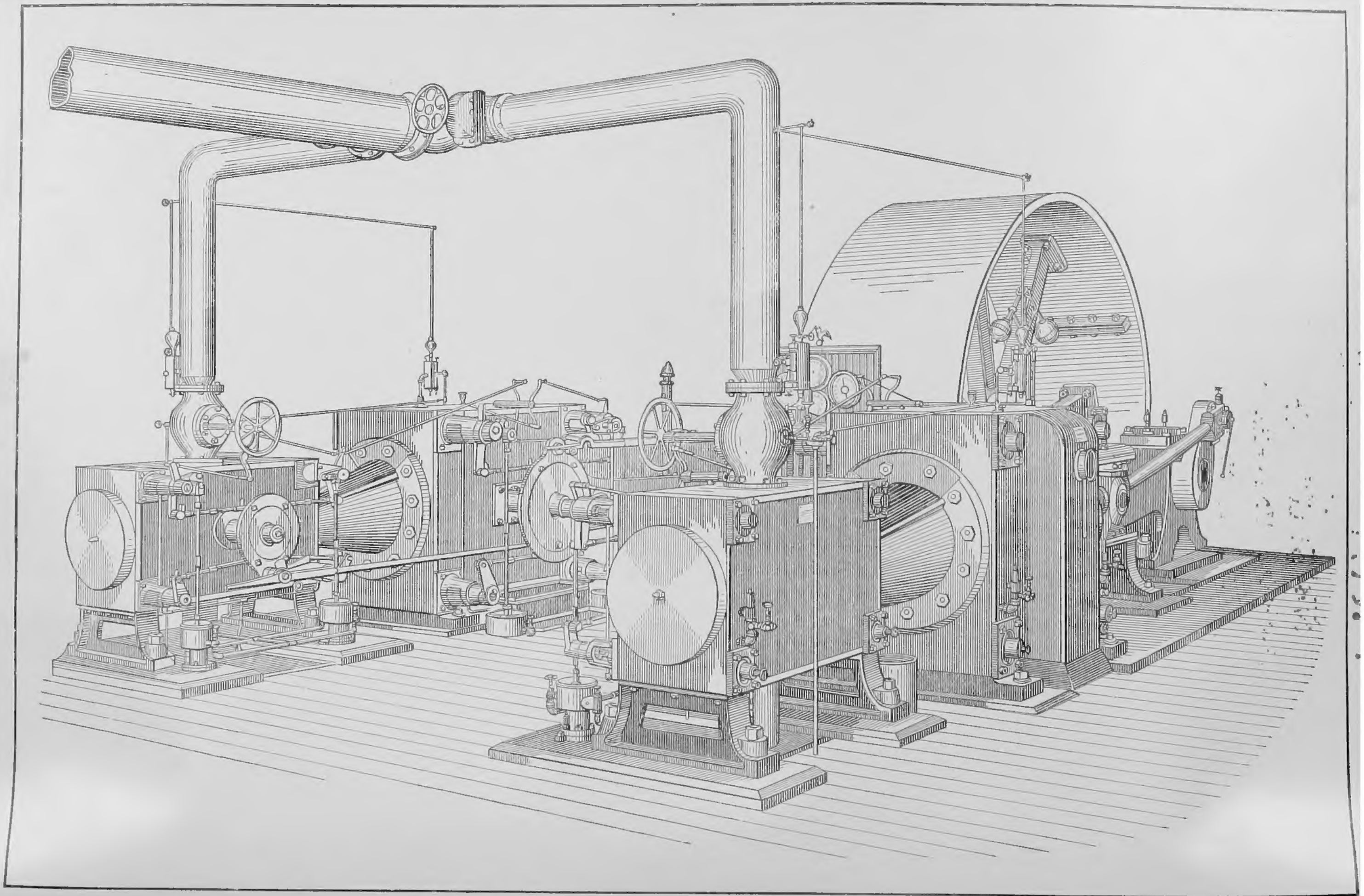
Cuts showing a few of the processes of the way rubber is treated in the manufacture of Hose, Packing, etc.

MODERN STEAM ENGINEERING

By GARDNER D. HISCOX, M.E.

INCLUDING AN ELECTRICAL SECTION

By NEWTON HARRISON, E.E.



Duplex Tandem Compound Corliss Condensing-Engine.

MODERN STEAM ENGINEERING

IN THEORY AND PRACTICE

A NEW, COMPLETE, AND PRACTICAL WORK FOR

Steam-Users, Electricians, Firemen, and Engineers

CONTAINING LATEST PRACTICAL INFORMATION ON

BOILERS AND THEIR ADJUNCTS; ECONOMY OF STEAM-MAKING AND ITS USE FROM THE FUEL TO THE CONDENSER, WITH ILLUSTRATED DETAILS OF STEAM ENGINE PARTS; SUPERHEATED STEAM, ITS USE AND ECONOMY; DETAILS OF SLIDE-VALVE AND HIGH-SPEED ENGINES; CORLISS, COMPOUND, AND TRIPLE-EFFECT ENGINES; THE STEAM-TURBINE AND ITS WORK; THE COST OF STEAM POWER, ITS APPLICATION AND OPERATION IN POWER PLANTS FOR ELECTRIC GENERATION, PUMPING, REFRIGERATION, AND ELEVATORS

OVER TWO HUNDRED QUESTIONS, WITH THEIR ANSWERS, LIKELY TO BE ASKED BY THE EXAMINING BOARDS ARE GIVEN, AS WELL AS FORTY TABLES OF THE PROPERTIES OF STEAM FOR POWER AND OTHER USES

BY

GARDNER D. HISCOX, M.E.

Author of "Gas, Gasoline, and Oil Engines," etc.

WITH CHAPTERS ON ELECTRICAL ENGINEERING BY

NEWTON HARRISON, E.E.

Author of "Electric Wiring, Diagrams, and Switchboards"

ILLUSTRATED BY OVER 400 SPECIALLY MADE ENGRAVINGS

NEW YORK

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PREFACE

IT has been the aim of the author in the production of this work to fully meet the wants of the student and engineer in all the practical requirements for obtaining a mastery in the application and use of steam for power and other purposes in the full range of its usefulness.

A further object has been to bring the mathematical side of Steam-Engineering into such practical conditions that the engineer or student may be able to grasp the whole subject with only ordinary arithmetical acquirements by means of the figured repetition of the formulas.

In the forty-two tables included will be found a ready reference, covering all conditions of the properties of steam and its application for the production of power, ratios, engine parts and proportions, most useful in the service now devolved upon the duties of a successful engineer.

Owing to the wide experience of the author, who well knows the points a book like this must cover to be of greatest service to the men for whom it is written, he has treated at length the subject of Superheated Steam and the practical operation of the Plain Slide- and Piston-Valves and their gear, the Corliss Valves and valve-gear, also the Triple- and Quadruple-Expansion Engine and the work of the Indicator, as well as the Steam-Turbine, which is now coming to the front as a power-producer.

The duties of an Engineer, who is entrusted with the management and use of Steam in a private or public capacity, are given, as well as chapters on Refrigeration-Plants, Elevators, and Electric-Light Plants.

Questions as asked by the Examining Board are included, as well as their answers, which will prove of greatest help to those preparing for and desiring to procure a License as a Steam-Engineer.

Much time has been spent and great care taken in the preparation of this work, and the author trusts that it will many times over compensate the reader for its perusal.

GARDNER D. HISCOX.

NOVEMBER, 1906.

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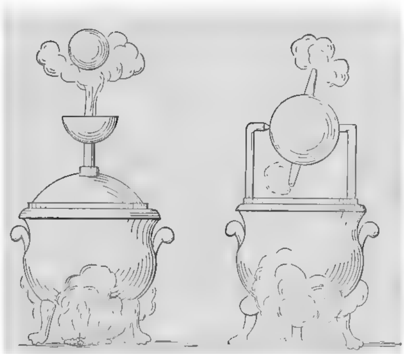
CHAPTER I

INTRODUCTION—HISTORICAL

STEAM has been known as a source of power since the earliest historic time.

It lifted the cover of the boiling-pot, even with a stone upon it, through the patriarchal ages, and later, with a tight-covered boiler, as designed by Heron of Alexandria, it became a source of power for motion in a rotary engine and in lifting a ball in a jet of steam, as here illustrated. Steam as a motive force appears to have been well known to the priesthood and magicians of Egypt as described in their incantations for creating awe and fear in the ignorant and superstitious people in that benighted age. There are reasons for believing that the expansive force of the steam that was evolved in heating the immense volumes of water for the hot baths at Rome, was employed to elevate and discharge the contents of the boilers; such being indicated by the investigations at Pompeii.



Force of the steam-jet. Heron's eolipile.

Steam was used in a feeble way by pressure and condensation for raising water during the first fifteen centuries of the Christian era, when its coming power only then began to enlighten the industrial horizon as the dawn of its brilliant day four hundred years later.

The experimental development of the properties and power of steam during the sixteenth century—the steam-played organ of Gerbert, the steam-gun of Leonardo da Vinci, the steam-boat of Blasco de Garay, the steam water-elevators of Baptist Porta—was a prog-

ress that to the acute mind of Roger Bacon opened a vista of the future which he expressed in the following prophetic words:

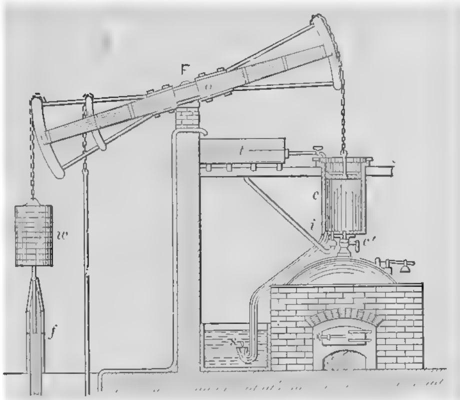
"Men may construct for the wants of navigation such machines that the greatest vessels, directed by a single man, shall cut through the rivers and seas with more rapidity than if they were propelled by rowers; chariots may be constructed which, without horses, shall run with immeasurable speed. Men may conceive machines which could bear the diver, without danger, to the depth of the waters. Men could invent a multitude of other engines and useful instruments, such as bridges that shall span the broadest rivers without any intermediate support. Art has its thunders, more terrible than those of heaven. A small quantity of matter produces a horrible explosion, accompanied by a bright light; and this may be repeated so as to destroy a city or entire battalions."

Destruction of Denys Papin's steam-boat in 1695, by the bargemen of the Seine (by Figuier).

Bacon was not a man to speak or write in this manner at random. His experiments led him to the conclusions he has thus recorded, for he was by far the most talented and indefatigable experimental philosopher of his age.

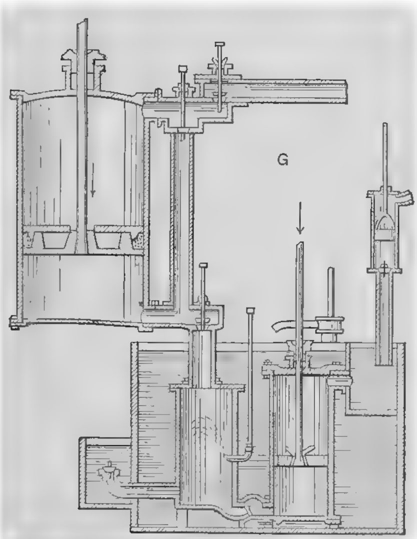
The first application of steam under pressure to the propulsion of a boat was made by Blasco de Garay at Barcelona, Spain, in 1543, although a few experiments on the power to lift water by steam-pressure were made at an earlier date and continued into the seventeenth century by De Caus, Branca, and the Marquis of Worcester. Dr. Denys Papin, in 1695, was probably the first to use the moving piston and the walking-beam on a steam-boat in the river Seine in France. To Dr. Papin may be attributed the origin of the steam-engine for power use. Steam under high pressure was used by him in the "Papin digester," a name surviving at the present time. Savory, a contemporary of Papin, in England, built water-raising

engines by direct action of steam and a vacuum; but little progress was made until Newcomen brought out the piston and walking-beam engine, for deep-well and mine pumping, in 1705; from which time there was but little improvement for a half century, until the time of James Watt, although Leupold, in 1720, invented a two-cylinder, single-acting piston-engine, moved by steam-pressure and exhausting into the atmosphere.



Newcomen's pumping-engine.

James Watt commenced experimental work on the steam-engine about 1761, making rapid progress in improvements of single-acting types, and by closing the top of the cylinder for the double-acting effect. The water-spray or separate condenser and air-pump, the atmospheric siphon condenser, the steam-jacketed cylinder, the parallel-motion crank, the fly-wheel, and the fly-ball governor were invented or applied by Watt previous to 1782, at which time he received a patent for the cut-off for using steam expansively in the cylinder. Thus it seems that the main features of the modern steam-engine were in use at the close of the eighteenth century.



Watt's single-acting condensing-engine.

Efforts to apply this pioneer of motive power to boats were made during the early part of the eighteenth century, and later in the century to vehicles, with a few improvements in its action and economy.

The compound steam-engine was patented in 1781 by Hornblower, in England, from which time steam-pressure as a practical power became progressive.

During the first century of the usefulness of steam little or no pressure was used in its operation for power, and not until the close of the eighteenth century was the then-called high-pressure engine brought into use, when 25 pounds per square inch was considered high pressure, and during the first half of the nineteenth century 50 pounds was named as high pressure, although much higher pressures were used for special purposes. In 1840 the Perkins steam-gun was operated by the author in New York City, with a steam-pressure of 1,000 pounds per square inch. It made wafers of bullets against an iron target; but the steam-gun did not prove practicable. At the dawn of the nineteenth century patents upon the principles of the application of high-pressure steam to engines were held by Trevethick and Vivian, in England, which were a menace to progress by contemporaries; yet progress in design and application to the propulsion of boats and the locomotive began the infancy of its future career.

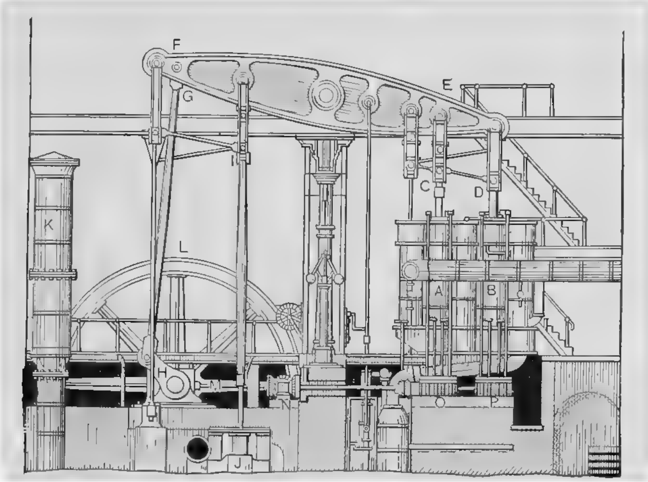
In the hands of Stevens and Fulton in the United States, and of Bell, Dodd, and others in England, steam navigation made a wonderful stride during the first half of the century; while the stationary engine plodded along seemingly in the rut of the slide-valve movement and slow speed. The cylindrical multitubular boiler became the leading type for the economical generation of steam and, with the internal furnace, the fixed type for marine and locomotive service.

The duty of a motive power is measured by the foot-pounds work produced per pound of a given heat-unit capacity of the fuel, or the initiative value of the power-producer. The progress of improvement during the first half of the nineteenth century was registered by the improvement in pumping service, which gradually advanced from a duty of 20,000,000 at the beginning of the century, to 108,000,000 in 1842, per bushel of 94 pounds Welsh coal, or equivalent to 1,148,936 foot-pounds per pound of coal; all due to improvements in boiler and engine design, compounding, and more perfect condensing effect.

During the latter half of the nineteenth century the duty in pumping-engines of the larger size had been raised to above 1,600,000

foot-pounds per pound of coal yielding 10,000 heat-units from the boiler in steam. This advance was largely due to increased boiler-pressure, 160 to 185 pounds gauge, and triple-expansion engines, giving efficiencies of above 21 per cent.

In this period the relative proportions of cylinder size and stroke have been changed to more nearly equalize the volume and wall surface, which means short stroke and larger diameter; types of the high-speed, tandem compound engines of to-day. From the middle of the century on, improvements in valve-gear continued to be made; the poppet-valve became established for engines in marine and river



Hornblower's compound pumping-engine.

service, and steam and exhaust lap and lead became an established principle in engines of the slide-valve and other types.

The latter half of the nineteenth century was a marked period in developing the efficiency and usefulness of the steam-engine.

Compression of the exhaust at the terminal of the piston-stroke became a fixed principle in design for smooth running in high-speed engines, although its efficiency is still a matter of discussion.

The quick and controllable valve-movement came with the Corliss type and established an advanced efficiency in the development of steam-power, and with increased steam-pressure, short cut-off and compounding have brought the coal value of a horse-power below 1 pound per hour.

The quadruple-expansion system seems to have reached a point that bars further progress in that direction; but with the opening of the twentieth century the long-dormant rotary principle received a new and practical impulse in the successful instalment of the steam-turbine; although not showing as yet an advance in steam efficiency, it fills a long-felt want for compactness and speed for marine and electric requirement, and thus has become the means for making a great advance in the usefulness of steam-power.

The principle of Heron's engine was the utilization of the reaction caused by the escape of steam from jets protruding tangentially from a hollow globe, this reaction causing the rotation of the globe.

More than seventeen hundred years later—in 1629—Giovanni Branca, an Italian inventor, devised an impact steam-turbine embodying the same principles as the familiar impact water-wheel of to-day, except that a jet of steam instead of water impinged upon the vanes of the paddle-wheel and caused it to revolve. The advent of the reciprocating steam-engine early in the eighteenth century diverted attention from the earlier attempts to perfect a rotating engine, and it was not until near the end of the nineteenth century that the steam-turbine again made its appearance as a commercial possibility. De Laval, in Sweden, in 1883, and Parsons, in England, in 1884, constructed successfully operating steam-turbines, and a continuous process of development and improvement has demonstrated the practicability and commercial value of this form of motor in two distinct types, obtaining efficiencies which rank with the best reciprocating engines. The performance of the steam-turbine, with the several very important advantages, justifies the belief that the field held for more than a century by the reciprocating engine of Watt is likely to be seriously invaded by this modern application of the earliest principles of steam-engineering, which is made possible by the better materials and workmanship and the more intelligent skill now available.

We cannot improve on the expressions of Prof. R. H. Thurston in regard to the progress in the realization of the practical possibilities and economics from the power of steam:

"The end of the nineteenth century is that of one which will always remain preëminent in history as the age in which the steam-engine

took shape in the hands of Watt and Sickles and Corliss and Greene, of Porter, and their successors, and thus brought in the factory system and all our modern methods of production, in the improvement of the condition of the people, and in all the material advancement in the industrial arts, which has made the century distinctively one of supremacy of the mechanic arts. The close of the century finds the steam-engine, though threatened with displacement by other motors, in the view of many writers, nevertheless the great motor of the age. Substantially all of the power employed by the civilized world is supplied by this great invention—congeries of inventions, rather—the product of a series of improvements, of an evolution effected during the hundred years or more just past. The limit to be possibly attained in its development and perfection will always remain a subject of intense interest to the profession and to the world.

"Reviewing the history of the growth of this form of steam-engine, it will be seen that its progress has illustrated that of the machine in all its forms, and that the steam pumping-engine gives the engineer a record of greater extent and of more representative character, as exemplifying the evolution of the machine, than does any other type.

"The twentieth century will very probably see a change in the curve of our lines, if not, in some respects, a decided halt or a reversed curvature, and it is perhaps even more probable that the field of the steam-engine will become greatly restricted by the introduction of other heat-motors, as well as by the general employment of electricity as a medium of extensive power-distribution from hydraulic and pneumatic prime movers.

"The steam-engine has now been so far perfected, and the practical limits of pressure are coming to be so nearly approached by steam-boiler constructors and users, that but little more can be expected of the designer; and even with the costlier types of engine, practically justifiable with exceptionally high costs of fuel, uninterrupted working, and low values of money, as in some instances with the steam pumping-engine, commercially practicable progress seems likely henceforth to prove very slow. These costly types of engine must necessarily have a comparatively narrow field. With the common case of moderate cost of fuel, intermittent duty, comparatively

high value of money in the business, or absolute scarcity with the buyer, gains seem likely hereafter to be rather in the direction of cheapened methods of construction and simplification of design."

The progress in the economy of fuel by increased steam-pressure in marine service during the past three-quarters of a century has been most marvellous, and, together with the improvements in construction of both engines and boilers, multiple expansion with surface condensation, has resulted in the saving of about 900 per cent,

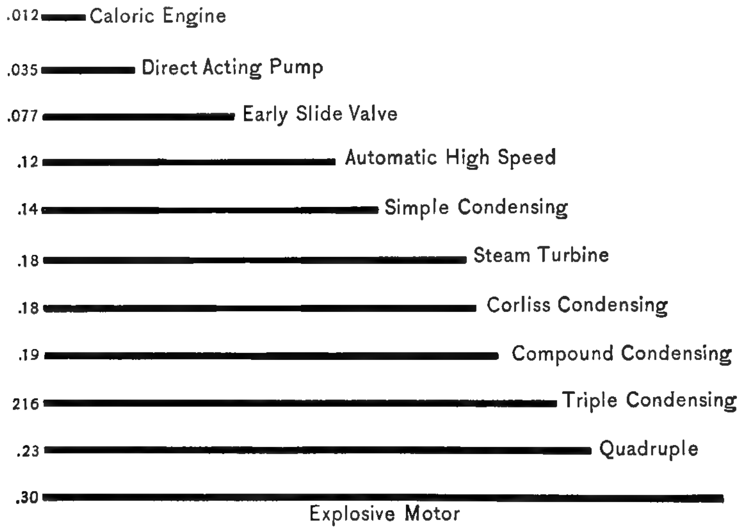


Diagram of progress.

reducing the old-time consumption of about 10 pounds to nearly 1 pound of coal per indicated horse-power. The progress in the rise of steam-pressure and consumption of coal per indicated horse-power, with few exceptions, is shown approximately in the following table:

Year.	Steam-pressure.	Coal per I. H. P.
1830	13 to 14 lbs. gauge	9 to 10 lbs. per hour
1840	18 " 25 " "	$5\frac{1}{2}$ " 6 " " "
1850	24 " 40 " "	4 " 5 " " "
1860	40 " 50 " "	3 " $3\frac{1}{2}$ " " "
1870	50 " 75 " "	$2\frac{1}{2}$ " 3 " " "
1880	75 " 100 " "	$2\frac{1}{4}$ " $2\frac{3}{4}$ " " "
1890	125 " 150 " "	$1\frac{1}{2}$ " 2 " " "
1900	160 " 200 " "	$1\frac{2}{5}$ " $1\frac{3}{10}$ " " "

In stationary service the steam-pressures have been greater than above stated in the earlier years, and the coal-saving has been improved in the most modern designs for the greatest possible expansion and mechanical efficiency for high-power service. The diagram of progress shows, in percentages, the approximate progress of thermal efficiency in different types and designs of engines for motive power.

The rapid progress recently made in steam-turbine design has given it a leading position in its special field of usefulness. Speed with power is a rare combination for useful effect in electrical generation, as well as in marine propulsion, in which both have made new records in their respective lines of practical operation.

We can scarcely realize the fact of the startling changes in the industrial and financial values in all the civilized world that have occurred within our memory and that have been due to education and its bearing upon this inventive age, and in which steam, with its work, with been one of the principal factors.

CHAPTER II

STEAM AND ITS PROPERTIES

STEAM, the vapor from water, is, like water, the product of a combination of the so-called permanent gases, hydrogen and oxygen, in the proportion by weight to one of the former to eight of the latter gas, and by volume one of hydrogen to two of oxygen.

This combination of these gases to form water, or its vapor, and steam is permanent up to a temperature to or above 2,000° F., when dissociation takes place from heat alone; but at much lower temperatures when in contact with ignited carbon in coal and other fuels; the oxygen combining with the carbon forming carbonic acid, CO_2, and carbonic oxide gas, CO, setting hydrogen free, and thus forming hydrocarbon compounds.

Water vaporizes at temperatures below its freezing-point, and as ice its vapor-pressure becomes zero at about −101° F. At 32° F. the vapor of water exceeds 208,000 volumes at as near a vacuum as practically possible, with an increasing density to about 20,000 volumes at 1 pound absolute pressure and temperature of 102° F., and to 2,361 volumes at 10 pounds absolute pressure and temperature of 193° F. As the temperature of the boiling-point is neared its density increases, and under atmospheric pressure (14.7 absolute), at 212° F. its vapor capacity is 1,646 volumes, or 26.36 cubic feet per pound of steam, weighing .03794 pound per cubic foot.

Steam when blown into the atmosphere expands to atmospheric pressure with a temperature of 212° F., and has but one-half the density of the atmosphere; hence it rises quickly, and, mixing with the air, is cooled, and by condensation into vesicles becomes a cloud. Pure steam is perfectly transparent, and so appears when looking through a jet close to the nozzle. The liberation of steam and vapor continues below atmospheric pressure in increasing volume per pound of water or vapor with a decreasing temperature of its boiling-point and absolute pressure. This property of evaporation at negative pressures and low temperatures has become a most valuable adjunct

of industrial work, indispensable in the modern methods of sugar and salt manufacture, and largely in use in the so-called vacuum-drying of various kinds of material and the condensation of liquids.

The following tables give the volume of 1 pound of water in vapor at various temperatures and pressures from and below the boiling-point of water at atmospheric pressure, and the elastic force of vapor at various temperatures.

TABLE I.—BOILING AND VAPORIZING TEMPERATURES OF WATER, AT AND BELOW ATMOSPHERIC PRESSURE, WITH PRESSURES AND THE VOLUME OF 1 POUND OF VAPOR. (Claudel.)

Temperature, Fahrenheit.	Pressure.		Volume of 1 pound, cubic feet.	Temperature, Fahrenheit.	Pressure.		Volume of 1 pound, cubic feet.
	Mercury, inches.	Per square inch, pounds.			Mercury, inches.	Per square inch, pounds.	
212°	29.92	14.70	27.2	120°	3.43	1.68	204.9
210	28.75	14.12	28.2	115	2.97	1.46	234.7
205	25.99	12.77	31.0	110	2.57	1.27	268.1
200	23.46	11.52	34.1	105	2.23	1.09	307.7
195	21.14	10.38	37.6	100	1.91	.94	353.4
190	19.00	9.33	41.5	95	1.64	.81	408.2
185	17.04	8.37	45.9	90	1.41	.69	471.7
180	15.29	7.51	50.8	85	1.20	.59	549.5
175	13.65	6.71	56.4	80	1.02	.50	641.0
170	12.18	5.98	62.4	75	.87	.43	746.3
165	10.84	5.33	69.8	70	.73	.36	877.2
160	9.63	4.73	75.0	65	.62	.30	1031.0
155	8.53	4.19	87.3	60	.51	.25	1220.0
150	7.55	3.71	97.8	55	.42	.21	1429.0
145	6.66	3.27	110.0	50	.36	.18	1695.0
140	5.86	2.88	124.1	45	.30	.15	2041.0
135	5.17	2.54	140.1	40	.25	.12	2439.0
130	4.51	2.21	158.7	35	.20	.10	2941.0
125	3.93	1.93	180.5	32	.18	.09	3226.0

TABLE II.—ELASTIC FORCE OF VAPOR OF WATER AT TEMPERATURES FROM 0 TO 212° F., AND ATMOSPHERIC PRESSURE OF 14.7 POUNDS. BAROMETER, 29.92.

Temperature, Fahrenheit.	Elastic force, inches, mercury.	Temperature, Fahrenheit.	Elastic force, inches, mercury.	Temperature, Fahrenheit.	Elastic force, inches, mercury.	Temperature, Fahrenheit.	Elastic force, inches, mercury.
0°	.044	62°	.556	122°	3.621	182°	15.960
12	.074	72	.785	132	4.752	192	19.828
22	.118	82	1.092	142	6.165	202	24.450
32	.181	92	1.501	152	7.930	212	29.921
42	.267	102	2.036	162	10.099		
52	.388	112	2.731	172	12.758		

Table II will be found convenient for aiding the formulas for computing the evaporation in Table IV for other air temperatures and humidities as stated in the heading of that table.

TABLE III.—BOILING-POINT OF PURE WATER AT PRESSURES BELOW THE ABSOLUTE ATMOSPHERIC PRESSURE OF 14.7 POUNDS PER SQUARE INCH.

Barometer, inches.	Absolute gauge-pressure, pounds.	Boiling-point, Fahrenheit.	Barometer, inches.	Absolute gauge-pressure, pounds.	Boiling-point, Fahrenheit.
29.92	14.70	212°	20.25	9.94	193°
29.33	14.40	211	19.82	9.73	192
28.75	14.11	210	19.41	9.53	191
28.18	13.83	209	19.00	9.33	190
27.61	13.55	208	18.59	9.12	189
27.06	13.28	207	18.19	8.93	188
26.52	13.02	206	17.81	8.74	187
25.99	12.76	205	17.42	8.55	186
25.46	12.50	204	17.05	8.36	185
24.94	12.23	203	16.31	8.00	182
24.44	12.00	202	14.27	7.00	174
23.94	11.75	201	12.23	6.00	166
23.45	11.51	200	10.19	5.00	157
22.97	11.28	199	8.16	4.00	147
22.49	11.04	198	6.09	3.00	135
22.03	10.81	197	4.07	2.00	123
21.57	10.59	196	2.04	1.00	109
21.13	10.37	195	0.00	0.00	98.7
20.68	10.15	194			

Pure water is said to boil in as near a perfect vacuum as its rising vapor by its rapid expansion will allow, at a temperature of 98° F. This was indicated by the double-bulb vacuum-tube in Franklin's experiment. The author has seen a carafe, partly filled with water at 50° F., placed under a nearly perfect vacuum made by the pump of a vacuum ice-making machine; the water boiled violently for a few seconds, the agitation not ceasing until ice began to form and became solid in a few minutes. The violent agitation was caused by the liberation of air.

Water holding salts and other substances in solution has its boiling temperature raised above 212° F., and thus becomes a valuable means of transmitting heat for boiling or concentrating liquids at open-air exposures and temperatures slightly above the boiling-point of water.

The following are convenient solutions for limiting temperatures in double-jacket kettles:

Common salt, for any temperature up to its point of saturation, 227° F.

Carbonate of soda, up to 220° F.
Nitrate of potash, up to 240° F.
Nitrate of soda, up to 250° F.
Carbonate of potash, up to 275° F.
Acetate of potash, up to 336° F.

The operation of chemical and industrial processes with heat above the boiling-point of water is a most important point for obtaining uniform results in the vast chemical, pharmaceutical, and provision-canning industries of this age. When absolute uniformity of temperature at a few degrees above the boiling-point of water is required, there is no more safe and reliable method than by the use of one of the above-named salts in part, as a bath or a saturated solution in open single-jacketed kettles heated by fire, or in double-jacketed kettles heated by steam, in which the boiling temperature of the heat-transmitting medium can always be under observation with a thermometer. By the direct heat of steam, as in the usual method of taking steam from a high-pressure factory-boiler, the pressure and temperature are often regulated by guess, a safety-valve, or a regulator; but these have their troubles and dangers.

TABLE IV.—APPROXIMATE HEAT REQUIRED FOR EVAPORATING WATER AT AND BELOW THE BOILING-POINT, FROM OPEN VESSELS IN CALM AIR AT A TEMPERATURE OF 52° F. AND 86 PER CENT HUMIDITY. (Box.)

Temperature of water.	Water evaporated per square foot per hour in calm air.		Time to evaporate 1 pound of water in hours.	Heat lost by radiation from surface. Units per hour.	Heat carried off by air. Units.	Latent heat of vaporization. Units.	Total heat to evaporate 1 pound of water. Units.	Total heat per square foot per hour. Units.	Cubic feet of air at 52° F. to evaporate 1 pound of water.
	Pounds.	Depth in inches.							
62°	.0143	.00275	70.0	11.3	888	1,071	2,750	39	4,807
72	.0343	.0066	29.2	23.4	753	1,064	2,500	86	2,036
82	.0615	.0118	16.3	35.2	649	1,057	2,280	140	1,160
92	.0986	.0190	10.2	47.0	555	1,050	2,080	204	747
102	.150	.0288	6.67	62.7	449	1,043	1,910	287	486
112	.221	.0425	4.52	76.7	387	1,036	1,770	392	350
122	.315	.0606	3.13	91.2	326	1,029	1,640	524	253
132	.454	.0873	2.20	106.8	278	1,022	1,535	698	184
142	.634	.122	1.58	122.5	241	1,015	1,450	918	146
152	.871	.168	1.15	141.1	206	1,008	1,376	1,197	112
162	1.18	.227	.848	156.4	193	1,000	1,326	1,564	95
172	1.57	.302	.637	175.5	179	993	1,284	2,016	81
182	2.06	.396	.485	193.2	168	986	1,248	2,573	70
192	2.66	.512	.374	215.7	164	979	1,224	3,268	64
202	3.41	.656	.293	237.7	161	972	1,203	4,106	58
212	4.32	.831	.232	257.0	160	966	1,186	5,112	54

The above table was derived from Regnault's experiments and verified by Box practically as to quantity, depth, and time. The formula for evaporation is (1) $E=(243+(3.7\times t)\times(V-v)$, in which E = evaporation per square foot per hour in grains; t = temperature of the water; V = elastic force of vapor at temperature t; v = force of vapor in air due to its percentage of humidity: or, by the table, II. $V=.388$ for 52° and $v=.388\times.86=.334$. For example, for water 62°, air 52°, and humidity .86,

(1) $E=(243+(3.7\times 62)\times(.556-334)=472.4\times.222=104.8$ grains, and

$\frac{104.8}{7,000}=.0149$ pound theoretical evaporation per hour.

The second and third columns in the table represent the experimental values.

The employment of a vacuum in boiling and evaporating water in the art of food preparation has become an important item in the industrial economy of these times. Among the many devices of the vacuum system of evaporation we illustrate the principles of its operation in an apparatus (shown in Fig. 9) for obtaining fresh water from salt water

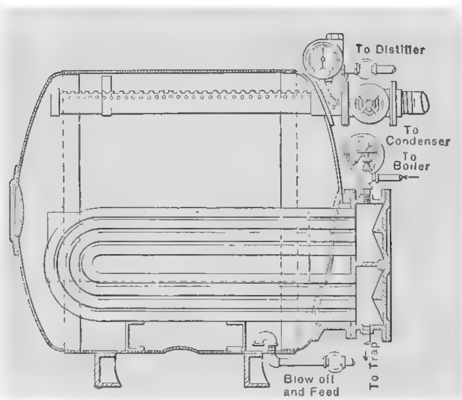


Fig. 9.—Fresh-water still.

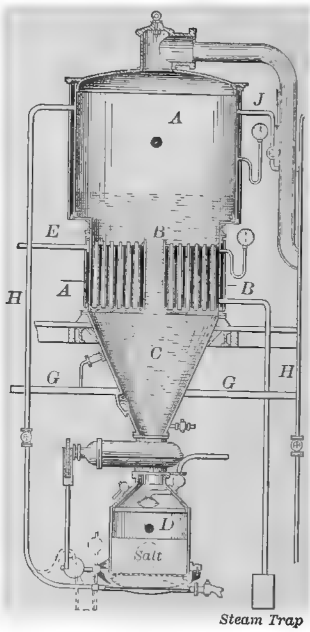


Fig. 10.—Vacuum salt-pan.

by the use of steam as the heating medium, which represents a fresh-water still:

The chamber is kept supplied half full of salt water and kept below saturation by blowing off. The vapor is drawn off through the perforated pipe at the top through a condenser by the vacuum-pump.

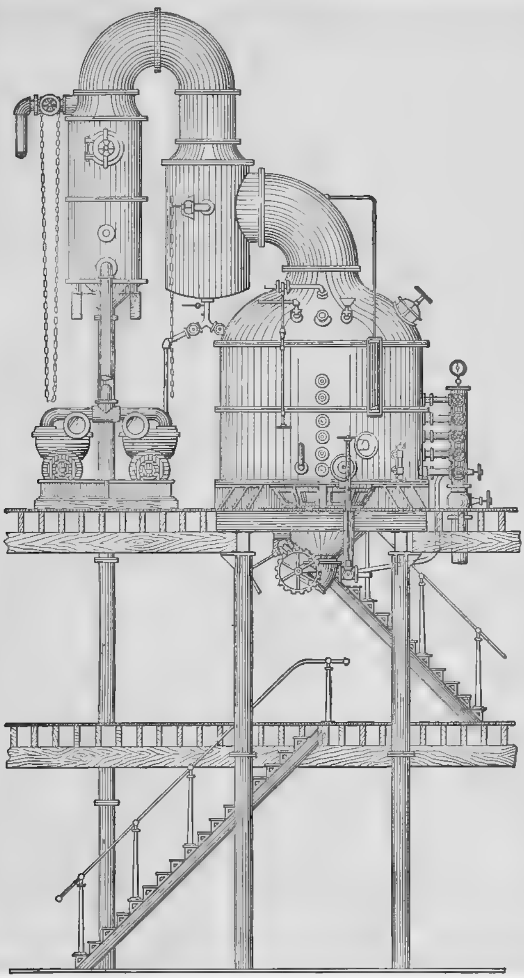


FIG. 11.—Elevation of a sugar-pan.

The boiling temperature of the salt water of the ocean is about 153° F., with a 26-inch vacuum. The condensed steam from the coils is saved and fed to the boilers. The condensed vapor from the salt

water is aerated and cooled for drinking. For the distillation of water for ice-making this principle seems to be the most economical conserver of heat known. By devices of double and triple effect, with coal at New York prices, pure distilled water can be produced at a cost of about 75 cents a thousand gallons; and when using the exhaust steam from a power-plant, the cost of producing a limited amount of distilled and aerated water is a mere nominal item.

The manufacture of salt by the vacuum process is becoming an important item in this industry. In Fig. 10 is shown the initial evaporating section of a triple-effect system consisting of three evaporating-pans set side by side with their terminal connected with the condenser and air-pump.

In this salt-making apparatus *A* is the vapor section, *B* the heating section, consisting of a series of vertical tubes connected from the boiler or the exhaust from an engine by the pipe *E*. In a triple-effect system the vapor-chamber *A* is connected with the heating section *B* of the next effect, and so on to the third effect, which has its chamber connected to the condenser and air-pump. *G* is the brine-inlet and *C* the crystallizing-chamber, from which the crystallized salt is discharged into the settling-chamber *D* through a slide-valve or gate. This is a continuous system and needs no suspension of the evaporating effect in each of the three sections as a triple effect.

In Fig. 11 is shown the elevation of one section of a sugar-boiling plant, operated on the dry system of evaporation, in which the vapor enters a jet-condenser and the condensed steam and water pass down a stand-pipe or siphon by gravity to a cistern 35 feet below the condenser, which seals the exit-pipe against atmospheric pressure. The air-pumps are only required to keep the system relieved of air and uncondensed vapor.

In this type of evaporator a series of copper coils enters the evaporating-pan from a header, and, circling around the inside, gives sufficient surface for the work of evaporation. Sometimes a surface condenser is used with a second pump for discharging the water of condensation.

CHAPTER III

GENERATION OF STEAM

FURNACES AND THEIR ADJUNCTS

THE economical generation of steam is becoming one of the most essential features in the world of engineering design for the production of power. The vast progress in manufacturing and producing industries of this age and their competitive relations not only require the utmost economy in the production of power, but even the fuel for power has its limit of production and its liability to increased cost; this serves as a warning to harbor our resources of the present for future emergencies.

The generator of steam and its power-developing agent, the steam-engine, have had a slow growth in development during the progressing ages of civilization, and only during the past century of scientific research have the principles for the economical application of this vast power been mathematically realized and applied.

The greatest economy in the production of steam begins at the furnace-door—the method of firing by which the most perfect combustion is produced and every atom of heat-producing fuel is consumed for the evolution of the highest temperature in the furnace.

Of the fuels in use for generating steam, anthracite and bituminous coal stand at the head, in accordance with their respective qualities in the carbon element. Wood, saw-mill refuse, lignite, peat, planing-mill shavings, bagasse, tan-bark, are in use with appropriate furnaces; while crude petroleum is gaining a leading fuel value in districts where it is cheaper than coal and in countries having small coal resources. The anthracite coals vary from 83 to 90 per cent in fixed carbon, with the volatile matter generally varying inversely to the fixed carbon, so that the total combustible averages about 90 per cent. With bituminous coals the volatile element is very large,

and with this perfectly consumed, its steam-making value is fully equal to anthracite per pound of combustible.

The steaming value of anthracite coal varies somewhat with its size for a given weight, as the ash products in merchantable coals vary from about $5\frac{1}{2}$ per cent in broken and egg size to above 16 per cent in pea and buckwheat sizes; yet with properly designed grates for burning the small sizes they are the most economical steam-makers from their low price.

Wood for steam-making is but little used, and is mostly derived from the waste in lumber-making, such as slabs, sawdust, mill-shavings, etc. Cord-wood, such as used for steam-making, contains about 55 per cent of its weight in combustible, combined with about 40 per cent of oxygen, which adds nothing to its heat-producing value, and only makes the wood highly inflammable. An average of $2\frac{1}{4}$ pounds of dry wood is equal to 1 pound of good anthracite or bituminous coal. In the following table are given the average weight of air-dried cord-wood per cord and its equivalent weight in anthracite or bituminous coal:

TABLE V.—AVERAGE WEIGHT OF AIR-DRIED CORD-WOOD AND ITS EQUIVALENT WEIGHT IN COAL.

KIND.	Pounds per cord.	Pounds of coal.
Shell-bark hickory	4,470	1,987
White oak	3,821	1,700
Red-heart hickory	3,705	1,646
Beach, red and black oak	3,250	1,444
Southern pine (pitch-pine)	3,375	1,500
Maple	2,878	1,279
Virginia pine	2,690	1,151
Spruce and New Jersey pine	2,137	949
White and yellow pine, hemlock	1,900	844

Lignite, or brown coal, is of recent geological formation. When dry it ignites easily and burns freely, and as a steam-producing fuel is between wood and coal as to bulk. Its specific gravity is from 1.10 to 1.25. It is used in many localities where it can be obtained at a less cost per combustible weight than coal or wood.

Peat or turf is of little value for steam-making. In a few localities, and in Germany, it is briquetted by drying and compressing, in which form it has been in use for some time, burning much like wood.

Coke is but little used for steam-making; it has a higher carbon value per pound than coal and makes a hot, clear fire.

Sawdust and shavings are used in wood-working mills for steam-fuel, more for the economy of disposing of them than for their steam-making value.

Bagasse and straw in the sugar and agricultural districts are used for steam-making as a means of disposing of by-products, and with bagasse fired in special furnaces under the long cylinder boilers set for this purpose, it seems to be a measure of real economy in turning the immense by-product of a sugar plantation to the best account.

The use of petroleum for steam-making has made a vast stride in late years, and especially in and near the oil districts, where its price competes with that of coal.

The oil constituents vary somewhat in different localities, with an average of carbon .86, hydrogen .13, oxygen .01, with their specific gravity varying from .80 to .94 and weighing from 6.6 to 7.6 pounds per gallon. The total heat of combustion varies from 19,000 to 22,000 heat-units per pound, with a theoretical evaporative power of from 19.6 to 22.7 pounds of water per pound of oil.

From comparison of the constituents of coal and petroleum the heating value of the oil is about 1½ times that of the coal; but in practice the heating value of oil has been found to be equal to twice the value of coal per pound in evaporating power. This has been accounted for by the more complete combustion of the liquid fuel and freedom of the tubes from soot and ashes. The admission of air being under complete control with the fine atomizing of the oil, brings the air and fuel into immediate and perfect contact with but a very small excess of air; while with coal much of the loss by combustion is due to excess of air. Another point in favor of oil fuel is the saving in banked fires and the convenience of instantaneous starting and extinguishing the furnace fire; while, with a well-constructed furnace and boiler-setting, the boiler will retain sufficient heat during the night for a quick morning start.

Natural gas, which is largely in use in the gas districts, with a heating capacity of from 900 to 1,200 thermal units per cubic foot, is a matter for economical consideration as to its cost per 1,000 cubic feet.

Mr. J. M. Whitham, who has investigated the natural-gas question, has summarized the subject in the following brief paragraphs:

1. In regard to burners, there is but little advantage possessed by one burner over another.

2. As good economy is made with a blue as with a white or straw flame, and no better.

3. Greater capacity may be made with a straw-white than with a blue flame.

4. An efficiency as high as from 72 to 75 per cent in the use of gas is seldom obtained under the most expert conditions.

5. The "air for dilution" is greater with gas than with coal, so that possible coal efficiencies are impossible with gas.

6. Don't expect, in good commercial practice, to get a boiler horse-power on less than from 43 to 45 cubic feet of natural gas, the same being referred to 60° F. and 4-ounce pressure above a barometer of 29.92 inches.

7. Fuel costs are the same under best conditions with natural gas at 10 cents per 1,000 cubic feet and semibituminous coal at $2.87 per 2,240 pounds. This is based on 3.5 pounds of wet coal being used per boiler horse-power per hour, or 45 cubic feet of natural gas.

8. Expressed otherwise, a long ton of semibituminous coal is the equivalent of 28,700 cubic feet of natural gas; while a short ton of such coal is the commercial equivalent of 25,625 cubic feet.

9. As compared with hand-firing with coal in a plant of 1,500 boiler horse-power output, coal being $2 per 2,240 pounds—considering labor-saving by the use of gas—natural gas should sell for about 10 cents per 1,000 cubic feet.

The largest constituent of natural gas is marsh-gas, CH_4, varying from 96 to 67 per cent; the next is hydrogen, H, varying from 22 to 6 per cent; the balance being ethane, C_2H_4, from 18 to 5 per cent, with traces of CO and CO_2 and free oxygen and nitrogen.

The economy of combustion in a boiler furnace is so much a matter of experience in the management of the fire and the relative volume of air allowed to pass through the furnace that is not required for combustion, and so much depends upon good practice derived from experience and good judgment in the handling of a fire, there is but little of value to be said in regard to its details.

In the disposition of the fuel elements by combustion, 1 pound of carbon requires 2.66 pounds of oxygen or 12 pounds of air or 162 cubic feet at boiler-room temperature for complete combustion, and having a heat value of 14,600 thermal units.

One pound of hydrogen requires 8 pounds of oxygen, $36\frac{1}{3}$ pounds of air, 490 cubic feet, for complete combustion, with a heating value of 62,000 thermal units. Thus, for example, for anthracite coal with carbon .91, hydrogen .028 per cent—$162 \times .91 = 147\frac{1}{2}$ cubic feet of air for the carbon, and $490 \times .028 = 13\frac{3}{4}$ cubic feet of air for the hydrogen, or 161 cubic feet of air per pound of coal for perfect combustion.

In practice perfect combustion cannot be obtained with less than from $1\frac{1}{2}$ to 2 times the quantity of air actually needed for perfect combustion.

The necessities for feeding the fire with open doors, and the faulty practice of letting too much air into the furnace for the consumption of the gaseous distillates above the fire, and for smoke consumption of bituminous fires, are serious drawbacks to the ultimate heat production of the furnace.

The volume of gases passing to the chimney is largely increased by excess of air to the furnace above the actual requirement for combustion, which for each pound of carbon in the coal and 50 per cent excess of air the products of combustion will increase in volume nearly in proportion to the excess of air, or to 18 pounds of air; and as the gases are expanded to nearly double their initial volume at the chimney temperature of 500°, it indicates a chimney volume of about 500 cubic feet of gases for each pound of carbon consumed in the furnace. Assuming that there is no air passing up the chimney other than that which has passed through the fire, the higher the temperature of the fire and the lower that of the escaping gases the better the economy, for the losses by the chimney gases will bear the same proportion to the heat generated by combustion as the temperature of those gases bears to the temperature of the fire. Then, if the temperature of the fire is 2,500°, and that of the chimney gases 500° above that of the atmosphere, the loss by the chimney will be $\frac{500}{2500} = 20$ per cent. As the temperature of the escaping gases cannot be brought down to that of the boiler, which is fixed, the temperature of the fire must be high in order to secure good economy.

BOILER-FURNACES, GRATE-BARS, AND MECHANICAL STOKERS

The boiler-furnace, its type and special design of firing appliances, are matters of no small moment in their contribution to the economy of steam-generation. Of the numerous models of shaking and tipping grates on the market, we can only illustrate examples of a few leading types.

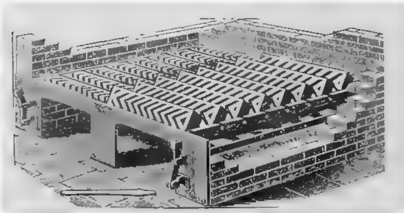


Fig. 12.—Tupper model.

The grate-bars in the Tupper furnace are in two sections so that one-half of the fire can be dressed at a time. The bars are placed crosswise and rock on trunnions by a hand-lever and connecting-bar. The diagonal spaces in the bars are made for any size coal required.

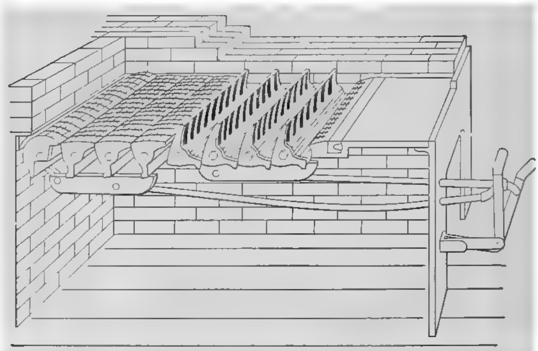


Fig. 13.—McClave shaking grate.

The McClave grate is made in fore and aft sections, so that by separate connections the front or rear section may be shaken or tipped for dumping the fire. Each bar forms a toothed comb with a bell-crank or stub-lever and connecting-rod, pivoted to all the bars in the section. When the bars are closed the whole grate is a continuous surface upon which a slicing-bar may be used.

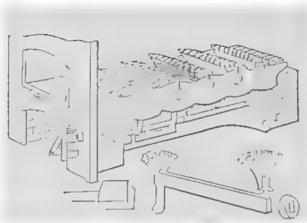


Fig. 14.—Sector grate-bar.

Fig. 14 shows a shaking and dumping grate composed of toothed sectors set astride pivoted crossbars with lever extensions, connected to transverse bars, so divided that the grate may be shaken in two or three sections. Heavy side-bars

carry the pivoted crossbars. The individual sectors can be readily removed when burned out and new ones inserted at trifling expense.

MECHANICAL STOKERS

The Roney mechanical stoker consists of a hopper for receiving the coal, a set of rocking, stepped grate-bars, inclined at an angle of 37° from the horizontal, and a dumping grate at the bottom of the

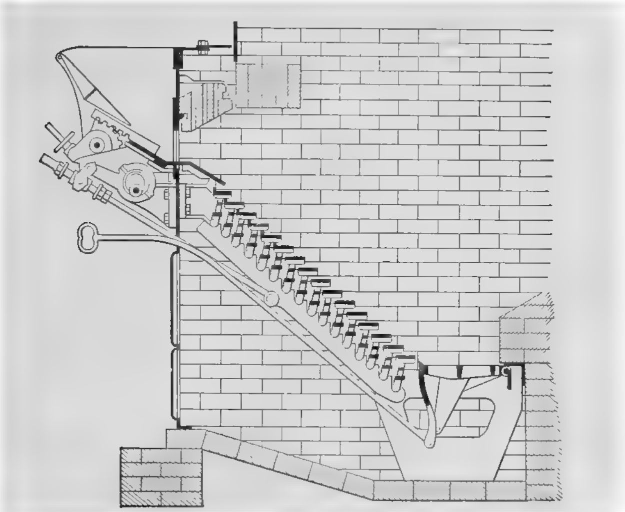


FIG. 15.—Roney mechanical stoker.

incline for receiving and discharging the ash and clinker. The dumping grate is divided into several parts for convenience in handling.

The coal is fed onto the inclined grate from the hopper by a reciprocating pusher, which is actuated by an agitator. The grate-bars rock through an arc of 30°, assuming alternately the stepped and the inclined position. The grate-bars receive their motion from a rocker-bar and connecting-rod, and these, with the pusher, are actuated by the agitator, which receives its motion through an eccentric from a shaft attached to the stoker-front, under the hopper. The range of motion of the pusher is regulated by the feed-wheel from no stroke to full stroke, and the amount of coal pushed into the furnace is adjusted according to the demand for steam. The motion of the grate-bars is similarly regulated and controlled by the position

of the lock-nuts on the connecting-rod. Each grate-bar is composed of two parts: A vertical web provided with trunnions at each end, which rests in seats in the side-bearers; and a fuel-plate, ribbed on its

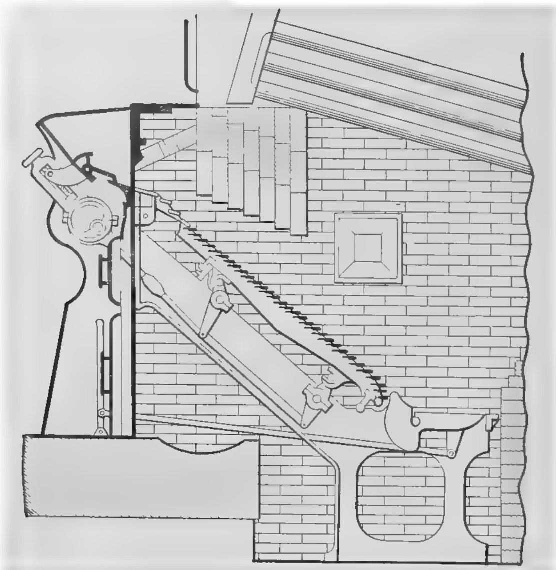


Fig. 16.—Acme stoker.

under side, which bolts to the web. These fuel-plates carry the bed of burning coal, and, being wearing parts, are made detachable, thus reducing the cost of repairs to the minimum. The webs are perforated with longitudinal slots, so placed that the condition of the fire can be seen at all times without opening the doors, and free access had to all parts of the grate to assist, when necessary, the removal of the clinker. These slots also serve an important purpose in furnishing an abundant supply of air for combustion.

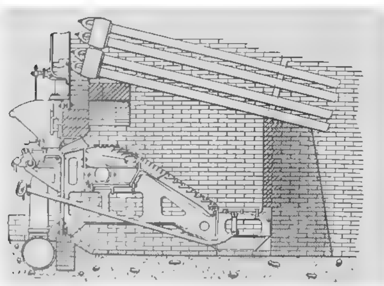


Fig. 17.—Soft-coal stoker.

The Columbian stoker is a special design for soft coal, which falls from the hopper into an upward inclined chute, and is pushed by a plunger onto a coking-plate and fixed small grate with an independent

air-feed chamber for regulating the coking. The pushing of the fresh coal against and under the coking-bank causes the coked coal to slide down the inclined grate. A supplementary rocking grate at the rear discharges the refuse. The air-feed to the small chamber at the top of the grate may be under pressure from a blower, and, thus mixing air with the smoke or distillates of the fresh coal, completes its combustion in the hot part of the furnace.

The travelling chain-grates are receiving much attention, and we illustrate the principles of several designs in this type of grate for soft coal.

In the Playford model a multilink-grate moved by a sprocket-shaft carries the coal, fed from a hopper, forward under the boiler; the grate returning over a drum at the bridge-wall. A screw-conveyer brings the ash and clinker forward to the pit.

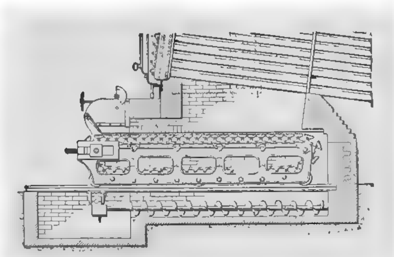


FIG. 18.—Playford stoker.

At the front of the furnace, immediately above the coking end of the grate-surface, is a fire-brick arch made of common sized fire-brick. The arch becomes highly heated, thus making the front end of the furnace a reverberatory chamber in which the gases are liberated from the coking fuel by distillation. The coked fuel in process of combustion is carried by the grate toward the rear of the furnace, while the ash and refuse are dumped automatically under the flat arch at the bridge-wall. Forming the top of that which otherwise would be an ordinary bridge-wall is a straight arch, overhanging the rear end of the grate-surface.

The Green travelling link-grate, which we illustrate in Fig. 19, is made up of long thin links of considerable depth and a large and uniform air-space around each link has been provided. The longer links afford an increased overhang, so as to shear any clinker which, during the travel of the grate, may be brought up to the bridge-wall, while at the same time it completely clears the ash from all the air-spaces of the chain at every turn around the rear sprockets. The links of the chain are connected together by bars of oval section,

which pass through round holes in the links or clips. The clips are engaged by the bars, and they in turn are locked and held in correct position by binding links at each end. The holes in the clips have a slot extending to the bottom edge, permitting any link or clip to be removed and replaced by another one without breaking the chain, removing the bars, or interfering with the service.

The chain is supported at frequent intervals by rolls extending under the entire width.

The framework is well braced and stiffened, to meet the requirements of hard service. In stokers exceeding 7½ feet in width there is

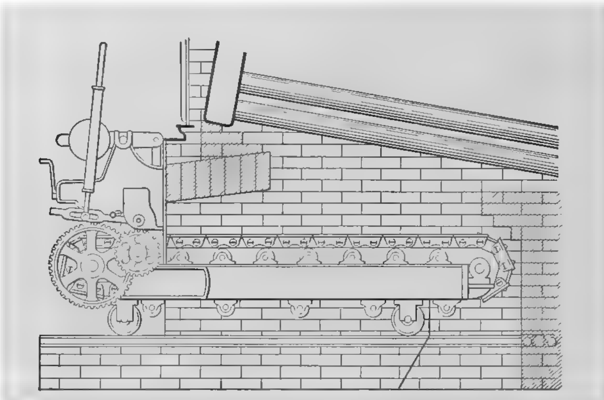


Fig. 19.—Green travelling link-grate.

provided an additional girder in the centre for the support of the upper roll-shafts.

The driving mechanism consists of an eccentric with a rod and lever, communicating motion by a ratchet and pawls to a train of gearing. This arrangement permits quick adjustment of speed of travel within a wide range. The gearing rests in a strongly braced frame. Steel pinions and gears are used throughout. The eccentric-connections from the shaft, placed either above or below, are made through a long relief-spring, thus preventing undue strain upon the gear-train or the chain in case of accident or stoppage.

A single large grate completely filling the entire furnace width is desirable, and more effective than two small grates occupying the same space. The flat coking-breast or ignition-arch combined with a chain-grate, as previously described, permits the successful use of

grates of any required width, and also obviates the further necessity of limiting the size of boiler-units because of inability to provide adequate grate-area.

In the American stoker the coal is carried under the grate from the hopper by a spiral screw and forced up over the grate. The screw-conveyer or worm (shown in the cut) is located in a trough that extends under the magazine, beneath which is the air-box under pressure from a blower.

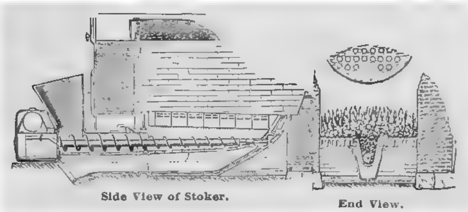


FIG. 20.—American stoker.

In operation the coal is fed into the hopper, and is carried by the conveyer into the magazine, which it fills, and, overflowing on both sides, spreads upon the sides of the grate. The coal is fed slowly and continuously, and, approaching the fire in its upward

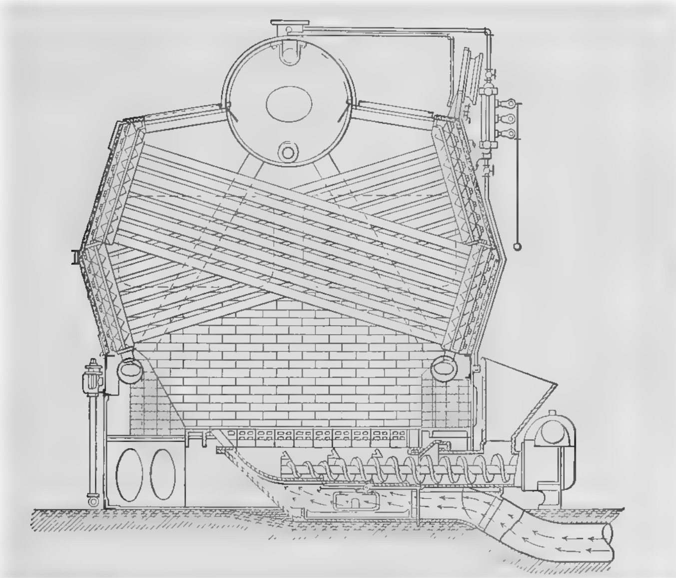


FIG. 21.—American stoker under a Worthington water-tube boiler.

course, it is slowly roasted and coked, and the gases released from it are taken up by the fresh air entering through the tuyeres, which

ignites them, and the coal is then delivered as coke on the grate above. The continuous feeding gives a breathing motion to this coke-bed, thus keeping it open and free for the circulation of air.

Every pound of coal fed into the hoppers passes through the gas-making process. The non-combustible matter is taken from the furnace in the shape of the usual ash. There is practically no soot. With these results it is obvious that the combustion must be extraordinarily good, resulting in a practically smokeless stack.

The finest of slack coal and also lump coal can be used, as any lump that can be fed into the hoppers will be crushed by the conveyer, there being provided a set of teeth, placed at the mouth of the conveyer, against which the coal is squeezed and broken.

The end-thrust of the conveyer is taken by a frictionless ball-bearing.

The conveyer-shaft is a $2\frac{3}{4}$-inch steel shaft, on which are strung what are called "flights." These "flights," by their reduced diameters, distribute the coal equally over the entire width and length of the furnace. The entire mass of coke above the tuyère-blocks and over the side-grate is ignited. The air enters the stoker from front or rear beneath the hopper, and discharges through the tuyère-openings. The discharge of air into each stoker is regulated by a wind-gate located at the mouth of the wind-chamber.

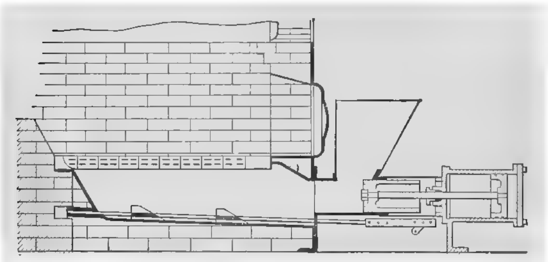


Fig. 22.—Jones stoker.

Air is supplied to the ash-pit by supplementary pipes from the main air-trunk.

The Jones stoker consists of a plunger which may be operated directly by a steam-piston, and which pushes a charge of coal, falling from the hopper onto the fore-plate of the grate, where it is coked, the smoke and gases being drawn into the hot fire and burned. These stokers are well adapted to other than boiler-furnaces and operated by levers in place of the steam-cylinders.

LIQUID FUEL

Of all forms of liquid fuel crude petroleum has become the standard for economy, and its increasing production has so decreased its cost for fuel that in many parts of this country and in other countries it is cheaper than other fuels.

The methods of its economical use have been so improved by trials and experience derived from the various detailed tests of the past few years, that its best results are now realized in very exact designs of furnace-construction.

Some of the advantages in the use of petroleum for steam-making are that its heating power is greater per pound than that of any solid fuel; that it permits of continuous firing in a closed furnace, free from draughts of cold air; that its combustion is complete, with no loss of heat by ashes, smoke, or soot; and that the quantity of heat required to maintain a constant pressure of steam may be controlled by the simple adjustment of a valve in the oil-supply pipe. The boiler-tubes are always clean and in the best condition for the transmission of heat to the water; starting and discontinuing of the fire are but the work of a moment, and all cleaning of ashes and *débris* avoided. The admission of air being under complete control, and the fuel being burned in atomized particles in contact with the air, only a small excess of air above that actually necessary for the complete combustion of the fuel being required; all these are points of economy.

The relative commercial value of coal and petroleum for steaming-fuel, apart from their convenience in operating a steam-plant, may be summed up in their relative heat-unit values. A net ton of coal is credited with 29,000,000 heat-units; a barrel of oil of 42 gallons, weighing 6.8 pounds per gallon, at 19,000 heat-units per pound, foots up to 5,426,400 heat-units, or $5\frac{1}{3}$ barrels to equal 1 ton of coal; so that at 90 cents per barrel for oil, it is equal in heat-unit value to coal at $4.80 per ton; but where coal is dear and oil as low as 50 cents per barrel, it is equal to coal at $2.60 per ton, and in locations where coal is about $6 per ton the difference in favor of oil is very apparent.

The burners or injector-nozles are made in a variety of forms; but all operate on the same principle. The steam-and-air method of atomizing the oil has not as yet become assured as to the use of

one or both combined, but so far the practice has favored steam as the atomizing agent, with an indraught of air by the force of the jet. A combined steam-and-air jet or an air-jet alone requires the use of an air-compressor, and if such is installed with independent power the steam can be left out of the process with the most economical results, for the steam requires a high temperature added before dissociation, and only then returns its absorbed heat by the reunion of its hydro-oxygen elements.

There is quite a wide-spread misconception regarding the part that the steam which is used for atomizing purposes plays in effecting combustion. It is supposed by many that after atomizing the oil the steam is decomposed and that the hydrogen and carbon are again united, thus producing heat and adding to the heat value of the fuel. While it may be true that the presence of steam may change the character and sequence of the chemical reaction, and result in the production of a higher temperature at some part of the flame, such an advantage will be offset by lower temperatures elsewhere between the grate and the base of the stack. All steam which enters the furnace will, if combustion is complete, pass up the stack as steam, also carrying with it a certain quantity of waste heat. The amount of this waste heat will depend upon the amount of steam and its temperature at the entrance of the furnace. The quantity of available heat, measured in thermal units, is undoubtedly diminished by the introduction of steam. In an efficient boiler it is quantity of heat rather than intensity that is wanted. For many manufacturing purposes intensity of heat may be of primary importance, but in a steam-generator a local intense heat is objectionable on other grounds than those of economy, viz., its liability to cause leaky tubes and seams from the unequal expansion of heating-surfaces.

In a series of naval official trials with crude-petroleum fuel, the following conclusions were arrived at:

"*a*. That oil can be burned in a very uniform manner.

"*b*. That the evaporative efficiency of nearly every kind of oil per pound of combustible is probably the same. While the crude oil may be rich in hydrocarbons, it also contains sulphur, so that, after refining, the distilled oil has probably the same calorific value as the crude product.

"*c*. That a marine steam-generator can be forced to even as high a degree with oil as with coal.

"*d*. That up to the present time no ill effects have been shown upon the boiler.

"*e*. That the firemen are disposed to favor oil, and therefore no impediment will be met in this respect.

"*f*. That the air requisite for combustion should be heated if possible before entering the furnace. Such action undoubtedly assists the gasification of the oil-product.

"*g*. That the oil should be heated so that it can be atomized more readily.

"*h*. That when using steam higher pressures are undoubtedly more advantageous than lower pressures for atomizing the oil.

"*i*. That under heavy forced-draught conditions, and particularly when steam is used, the board has not yet found it possible to prevent smoke from issuing from the stack, although all connected with the tests made special efforts to secure complete combustion. Particularly for naval purposes is it desirable that the smoke nuisance be eradicated, in order that the presence of a warship may not be detected from this cause. As there has been a tendency of late years to force the boilers of industrial plants, the inability to prevent the smoke nuisance under forced-draught conditions may have an important influence upon the increased use of liquid fuel.

"*j*. That the consumption of liquid fuel cannot probably be forced to as great an extent with steam as the atomizing agent as when compressed air is used for this purpose. This is probably due to the fact that the air used for atomizing purposes, after entering the furnace, supplies oxygen for the combustible, while in the case of steam the rarefied vapor simply displaces air that is needed to complete combustion.

"*k*. That the efficiency of oil-fuel plants will be greatly dependent upon the general character of the installation of auxiliaries and fittings, and therefore the work should only be intrusted to those who have given careful study to the matter, and who have had extended experience in burning the crude product. The form of the burner will play a very small part in increasing the use of crude petroleum. The method and character of the installation will count for much, but where burners are simple in design and are constructed in ac-

cordance with scientific principles there will be very little difference in their efficiency."

It may be further said that the heat of steam at high boiler-pressure adds greatly to the atomizing effect by heating the oil in its passage through the injector and thus vaporizing its more volatile constituents; whereas the cooling effect, by the expansion of compressed air, would have an opposite effect, unless the compressed air could be highly heated by passing through a coil in the smoke-chamber.

PETROLEUM-OIL BURNERS

In Fig. 23 we illustrate an oil- and air-burner in which the oil enters by the pipe A to the central nozle with its regulating-valve C. The compressed air enters through the pipe B and issues through an annular nozle, and is retained by an outer nozle which may be more or less extended for a thorough atomization of the oil.

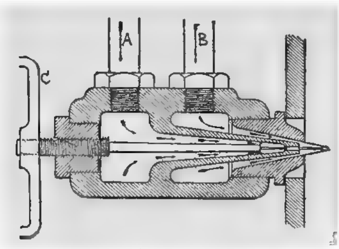


Fig. 23.—Oil- and air-burner.

A burner of English origin is shown in Fig. 24, for the combined use of oil, steam, and air, which are combined in an expanding nozle. The oil enters by the rear pipe, and its flow is regulated by a needle-valve. Steam enters by the middle pipe, forming an annular jet around the oil, while

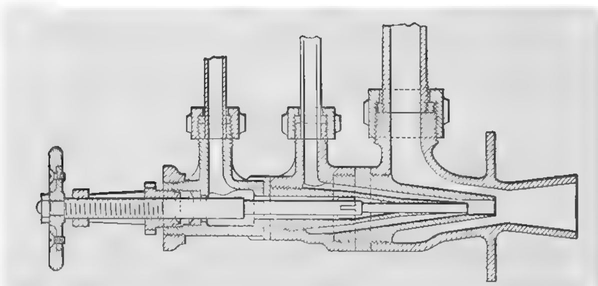


Fig. 24.—Oil-, steam-, and air-burner.

the air enters by the large pipe and forms an annular jet surrounding both oil and steam as the combination enters the expanding nozle.

In Fig. 25 is shown, in plan and section, an oil-burner used on the locomotives of the Southern Pacific railroads.

It is a combined oil-, steam-, and air-burner with a wide, thin mouth and chambers closed at the sides, so that the compartments are

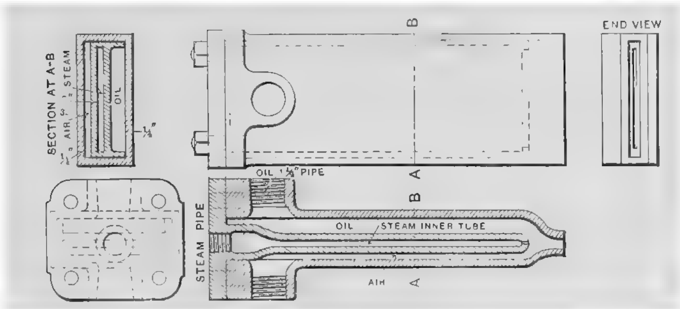


FIG. 25.—Flat-nozle burner.

separated in tiers, one above the other. Oil flows along the flat top chamber, with the steam in the central chamber to heat the oil, and the oil, steam, and air meet at the end of the inner nozle, where the oil is atomized in contact with the air and steam and projected through the end nozle.

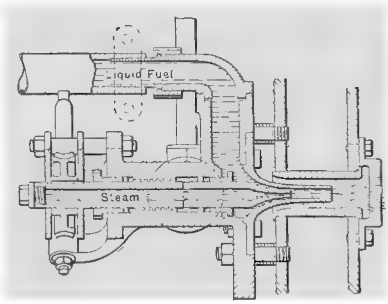


FIG. 26.—Urquhart burner.

In Fig. 26 is illustrated the Urquhart type of burner, in use on the Russian railways. It is a combined oil-, steam-, and air-burner. Steam enters the hollow valve-spindle through side-ports, while the oil enters by a side-pipe and through an annular aperture outside of the steam, the nozle of which extends into a hollow stud between the plates of the boiler-front. A plate held off from the boiler carries the injector with an air-space to give air to the jet in the stay-tube.

The Oil City burner (Fig. 27) is of the oil and steam combination type, with a cap over the end for inducing a draught of air around the oil- and steam-jet. The nozle is made long that the steam on the outside of the oil-nozle heats the oil for more perfect atomization. The oil- and steam-flow are both adjustable by valve-wheels.

The F. M. Reed burner (Fig. 28) is somewhat novel in the manner of combining the oil, steam, and air.

The oil-flow, regulated by an outside valve, enters through the

valve-spindle, surrounded by steam from a side entrance, both issuing through a short expanding nozle into a chambered nozle, which

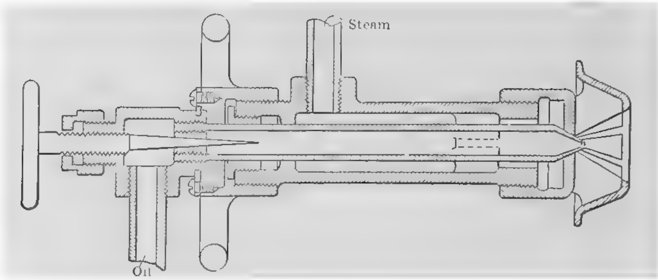


FIG. 27.—Oil City burner.

completes the atomizing of the oil; outside of this is a larger chambered nozle through which air is drawn by the force of the jet, and, mixing

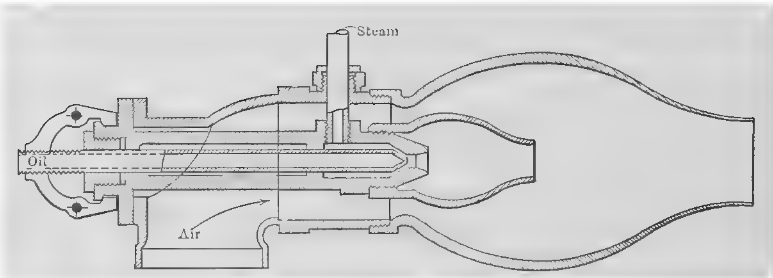


FIG. 28.—F. M. Reed burner.

with the atomized oil and steam, the final issue to the furnace is ready for complete combustion.

CHAPTER IV

TYPES OF BOILERS

One of the earliest types of the modern water-tube boiler was made in 1804 by John Stevens at Hoboken, N. J., and with its twin-screw propellers is now preserved in the Stevens Institute as the century memento of our present water-tube boilers and of the principles of twin-screw propulsion. After forty years the same boiler, engine, and screws were placed in a boat of the original dimensions and propelled at the rate of eight miles per hour.

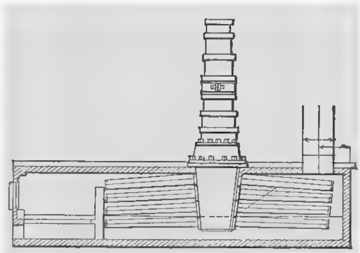


Fig. 29.—Stevens boiler.

Boilers of the shell type continued in use as standard form during the nineteenth century, with an occasional interpolation of a water-tube boiler, a flash-boiler, and inter-circulating coil-boilers, as, for example, the Perkins type, which carried steam at 1,000 pounds pressure per square inch and was applied to a steam-gun which was exhibited in New York in 1839 and operated by the author.

The most simple form of boiler is the long, plain shell now much used on sugar-plantations, and under which bagasse is used for fuel.

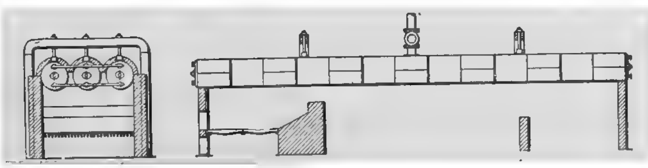


Fig. 30.—Cylinder boiler.

These boilers are usually set in nests hanging from beams supported by the side walls, and are closed in at their half-diameter. Their boiler-power may be computed at one-half their shell-area divided by 10 per horse-power.

The double-flue, cylindrical shell-boiler was a favorite form with wood-burning furnaces for steam-boats and wood-working mills,

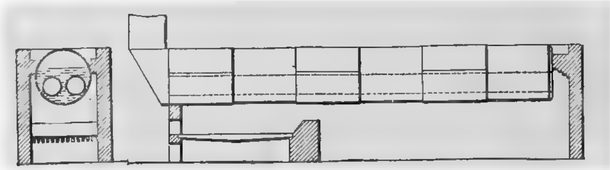


FIG. 31.—Double-flue boiler.

and is still in use in modified forms. One-half the shell- and all the flue-surface, divided by 11, equal the boiler horse-power.

The internally fired flue-boiler is now becoming antiquated from its limited furnace capacity, and has finally merged into the corrugated tubular furnaces of the marine type.

The horizontal tubular boiler has been a standard type for three-quarters of a century, and is yet the leading steam-maker within the range of its allotted pressure. For ease of care and convenience of repair it stands at the head of the list in number in use in all countries. When well proportioned for its work its economy is unchallenged, and at pressures of 100 pounds, and under, it is, when well constructed and cared for, as safe from rupture as other types.

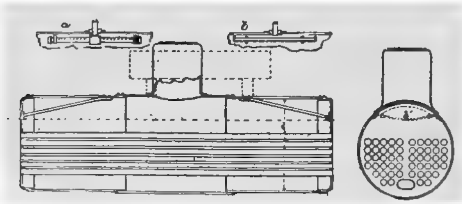


FIG. 32.—Cylindrical tubular boiler.

One-half the shell- and all the outside tube-surface, divided by 14, equal the boiler horse-power.

The Galloway boiler, an English type, has a cylindrical shell with an oval flue and is internally fired. It has two furnaces which merge into a combustion-chamber at the rear. This chamber is fitted with

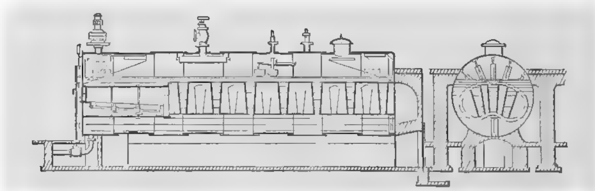


FIG. 33.—Galloway boiler.

tapered water-tubes for the purpose of increasing the effective heating-surface of the boiler and of promoting a better circulation of water; they also act as stays, largely increasing the strength of the flue to which they are fitted.

The heated gases after passing through the internal combustion-chamber return along the outside of the shell to the front and again to the rear end and to the chimney. It is considered an efficient

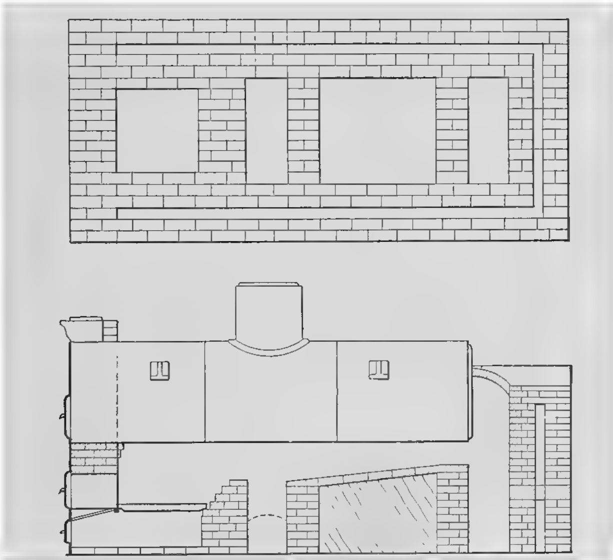


FIG. 34.—Plan and section of boiler-setting.

boiler. All the internal surface of flue and tubes and the shell exposed to heat, divided by 12, equal the boiler horse-power.

The setting and care of a cylindrical tubular boiler are matters of careful consideration. Air-leaks in the brickwork adulterate the hot gases of combustion and are the cause of fuel loss; so that too much pains cannot be taken in making the joints as close as the bricks will allow and fully flushing every joint with mortar.

Fig. 34 shows the plan and elevation of a flush-front boiler-setting with a filled-in rear chamber and recesses for catching the light ashes that pass over the bridge-wall. The air-space in the walls is shown in the cut much wider than needed, as 2 inches is wide enough for the largest boilers.

The filling of the rear chamber is of doubtful value, as the large area of this chamber serves as a setting-place for the light ashes that

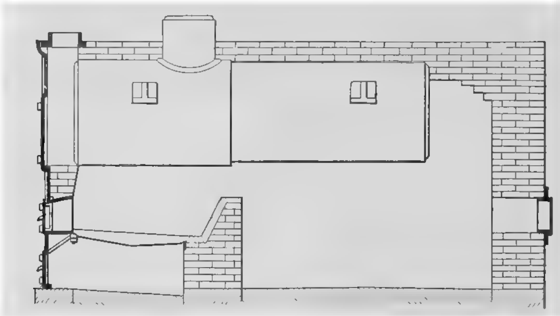


FIG. 35.—Section with flush front, open chamber.

are carried over the bridge-wall, and the large volume of hot gases and ashes is a strong radiant of heat to the rear end of the boiler.

In Fig. 35 is a section with solid brick walls carried above the top of the boiler and with an extension of the grate-surface onto the bridge-wall and a support of the back-chamber closure by a T beam.

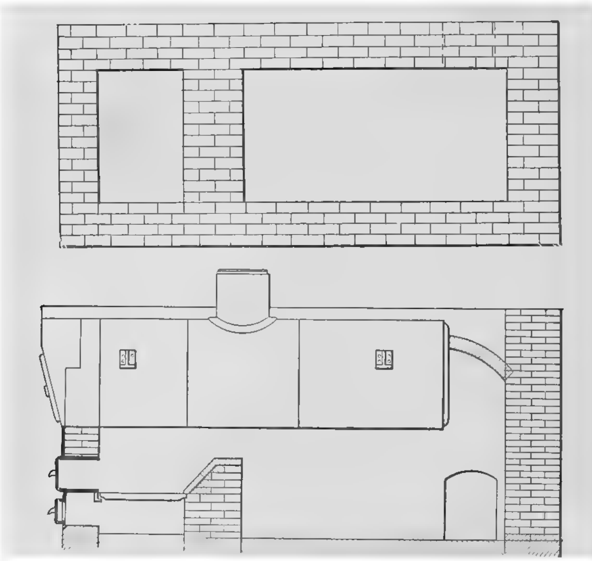


FIG. 36.—Plan and section of overhang boiler-front setting.

Domes on this type of boiler are not recommended, because they are a source of weakness; but they have their advocates on the plea

that they are steam-reservoirs and promoters of dry steam. A trifle larger shell and a dry pipe at the same cost is the safe and preferable plan.

Fig. 36 shows the plan and section of the setting of a boiler with an overhang front smoke-chamber, which rests on a half-front frame.

THE ROBB–MUMFORD BOILER

This internal-fired cylindrical boiler is a recent type of construction, with an internal fire-box of the Morrison corrugated type and with its full area of head filled with tubes; the lower shell is inclined, as is also the steam-drum above, in order to facilitate water-circulation. The design is very compact and enclosed in a brick-lined metal casing. It appears to be a good steamer. The fire-surface of the tubes, fire-box, and all of the shell exposed to heat, divided by 12, equal the boiler horse-power.

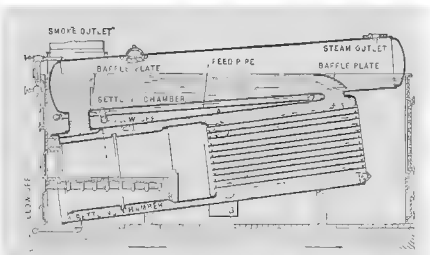


FIG. 37.—Internal fired, cylindrical tubular boiler.

The "Continental type" of marine boiler has a double corrugated tubular furnace with cylindrical shell and return-tubes. This form of construction is the general one for marine use, and is made in the large units with multiple furnaces. It is a modification of the "Scotch boiler" and is also made with double ends, set back to back with common or separate combustion-chambers.

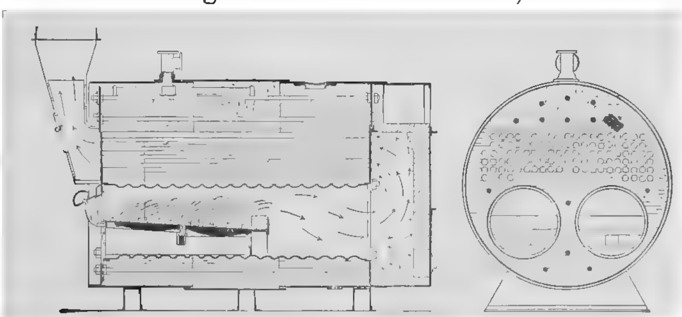


FIG. 38.—Marine boiler.

The down-draught system of combustion as applied to a Hein boiler is shown in Fig. 39, in which is illustrated the Hawley furnace; *c* is a tubular grate; *d*, tube-connection between the grate-header and front drum; *b*, tube-connection between the grate-header and rear drum, with blow-off.

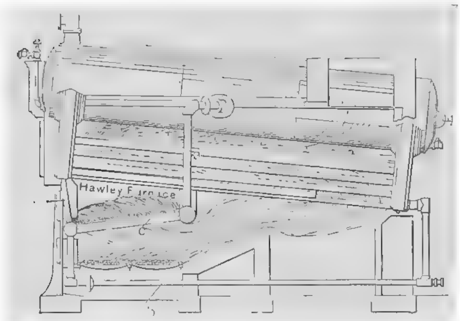


Fig. 39.—Down-draught furnace.

The Hein boiler has its water-legs in form of a wrought-iron chamber flanged and riveted to the steam- and water-drums, which are stayed throughout their breadth. The tubes are expanded in the heads, with a hand-hole plate opposite each tube for cleaning and repairs. Their steaming capacity is about 11 feet of heating-surface per boiler horse-power.

Among the many pipe-boilers on the market for special purposes and claims for efficiency and high-pressure safety, we illustrate the Herreshoff boiler (Fig. 40), which has gained much credit in the steam-yacht service. The inner coil is the evaporator and receives the feed-water through the heater-coil in the smoke-chamber. The conical coil at the top also acts as a heating-chamber and feeds the inner coil, while the outside coil is the superheater.

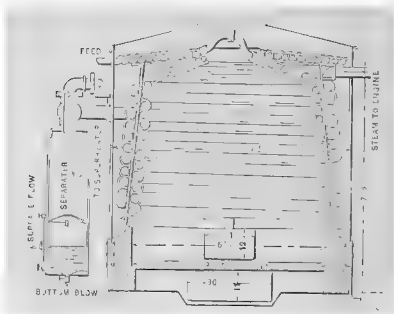


Fig. 40.—Herreshoff boiler.

A separator-drum at the side entraps any superfluous water that may be fed to the boiler, and also acts as a separator, giving to the superheater dry steam.

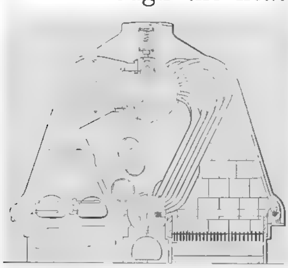


Fig. 41.—Thornycroft boiler.

The Thornycroft boiler (Fig. 41) is shown as a type in principle of a number of water-tube boilers on the market in Europe and the United States. It consists of a large steam-drum above and a water-drum below, con-

nected with a large number of bent tubes. The water-return is through a large tube at the rear end of the boiler. The same principle of action, with different designs in construction, is carried out in the Yarrow, the Moyes, and the See water-tube boilers, with straight tubes; the Boyer water-tube boiler, with return bend coils, and the Meehan and Sterling water-tube boilers, with bent tubes. The Du Temple marine water-tube boiler (Fig. 42), made in France, is of the Thornycroft type and is here illustrated. Although patented in 1876 it has all the essential qualities of the later water-tube boilers. A quick and powerful steamer, it has been much in use in the torpedo-boat service. Ample circulation is provided for by the direct back connections between the steam- and water-drums. The cut shows a half-section of each side.

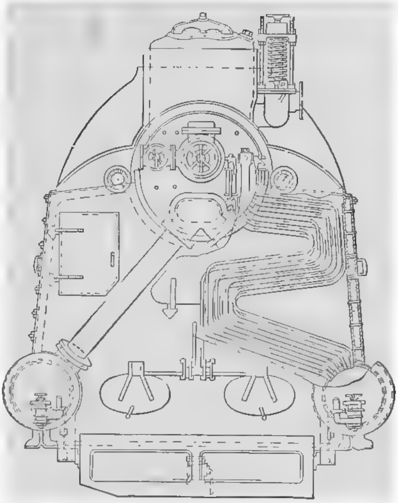


FIG. 42.—Du Temple boiler.

We do not deem it expedient as the object of this work to illustrate the large number of boilers on the market, whose builders claim merit and public favor for their special designs; nor can we go into the merits of their tests of economy and usefulness with any degree of safe judgment; rather let their work with users tell the story of their worth.

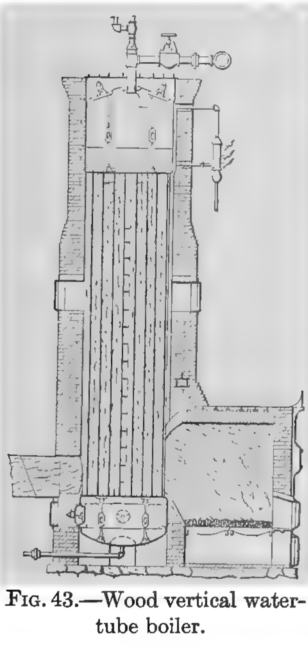


FIG. 43.—Wood vertical water-tube boiler.

Among the vertical water-tube boilers with straight tubes between the steam- and water-drums with outside furnaces we illustrate the Wood boiler (Fig. 43), which has large steam- and water-drums with

stayed heads; the tubes are arranged in front and back sections with a brick or tile partition carried up from the lower head to near the top, so that the products of combustion traverse the whole length of the tubes up and down and out to the chimney at the rear. The arched furnace projects at the front, and the whole boiler is enclosed in brick walls. Circulation is obtained by the more active upward current, with its steam in the fire section and induced down-flow in the rear section.

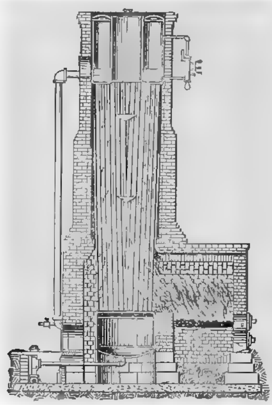


FIG. 44.—Cahall boiler.

Another design is the Cahall water-tube boiler (Fig. 44) with straight, nearly vertical tubes between an annular steam-drum at the top, through which the smoke passes, and a water-drum at the bottom. The furnace and combustion-chamber project outside at the front, and a circulating-pipe is carried from the steam-drum to the water-drum outside of the brick setting.

In Fig. 45 is shown the duplex water-tube boiler made by the Philadelphia Engineering Works. This boiler has straight vertical tubes between a pair of steam- and water-drums. The steam-drums are connected with several cross-pipes, as shown in the cut, and the water-drums are connected by a single neck. A brick wall between the two stacks of tubes directs the products of combustion upward through the tube-stack next to the furnace and down the opposite stack and to the chimney. The water-circulation takes the same course through the tube-stacks and drum-connections.

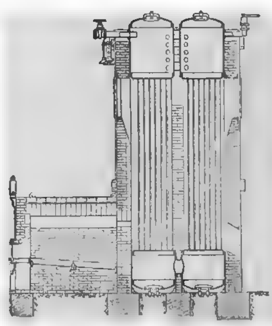


FIG. 45.—Duplex boiler.

In Fig. 46 is a section of the Sterling water-tube boiler, which consists of three upper or steam-drums and one lower or mud-drum, with the tube-ends bent so that all of them can properly enter the drums.

The steam-spaces of all the upper drums are connected, while the water-spaces of only the front and rear drums communicate. The

drums are made of flange steel, while the tubes are lapwelded steel, tested at 1,500 pounds pressure per square inch. These drums and the tubes form the boiler proper.

It is designed to be a safety-boiler, and the absence of flat surfaces renders stay-bolts and braces unnecessary; it will be seen that a

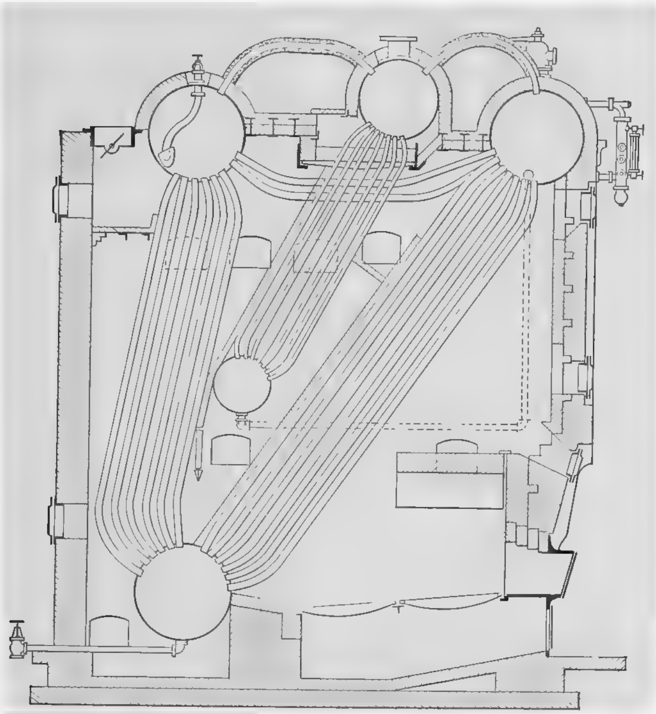


FIG. 46.—Sterling water-tube boiler.

fire-brick arch is built over the grates and immediately in front of the first section of tubes. This arch absorbs heat from the fire on the grates and becomes an incandescent, radiating surface. Tile partitions are so arranged that the hot gases pass the entire length of the three stacks of tubes.

The feed-water enters the rear upper drum, the coolest part of the boiler, and as it descends to the mud-drum is gradually heated by the gases, passing to the chimney to a sufficient extent to render insoluble much of the sediment that it contains, which is deposited in the mud-drum in the form of mud or sludge, from which it may be blown

off. The mud-drum is protected from the intense heat of the furnace by an ample bridge-wall, and acts as a settling-chamber.

The middle drum is connected to a supplementary drum by a series of tubes in the same manner as the others, and receives the priming, which, falling through the tubes to the lower drum, is reconverted into steam and superheated.

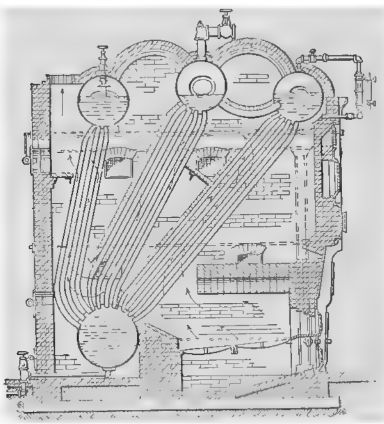


Fig. 47.—Sterling water-tube boiler.

By suitably disposed fire-tile partitions or baffle-walls, the gases from the furnace are led first up among the first bank of tubes, depending from the front drum, thence down the middle bank, thence up the rear bank, and on into the chimney at a reduced temperature.

In this long and circuitous passage the gases come in contact with all the tubes, which method insures a more or less complete delivery of their heat to the water.

In Fig. 48 is illustrated the latest style of the Babcock & Wilcox water-tube boilers, the most compact and economical design of all of their extensive manufacture, and best suited for generating high-pressure steam.

The vertical header style has the same general features of construction as their other styles, with the exception of having the tube-sheet side of the header "stepped" so that the header may be placed at right angles to the drum, instead of having it inclined, as in previous designs. This form permits of a shorter brick setting, thereby reducing the cost of erection and the floor-space occupied.

The last step in the development of the water-tube boiler, beyond which it seems almost impossible for science and skill to go, consists in making *all parts of the boiler of wrought steel,* including the sinuous headers, the cross-boxes, and the nozles on the drum. This was demanded to answer the laws of some of the Continental nations, and the Babcock & Wilcox Co. have at the present time a plant turning out forgings, as a regular business, which have been pronounced to be "a perfect triumph of the forgers' art."

One of the important points in the generation of steam is that it

should be dry as it leaves the boiler; and in this class of boilers the large disengaging surface of the water in the drum, together with the fact that the steam is delivered at one end and taken out at the other, secures a thorough separation of the steam from the water, even when the boiler is forced to its utmost. Most tubular, locomotive, and sectional boilers make wet steam, "priming" or "foaming" as it is called, and in many "superheating surface" is provided to "dry

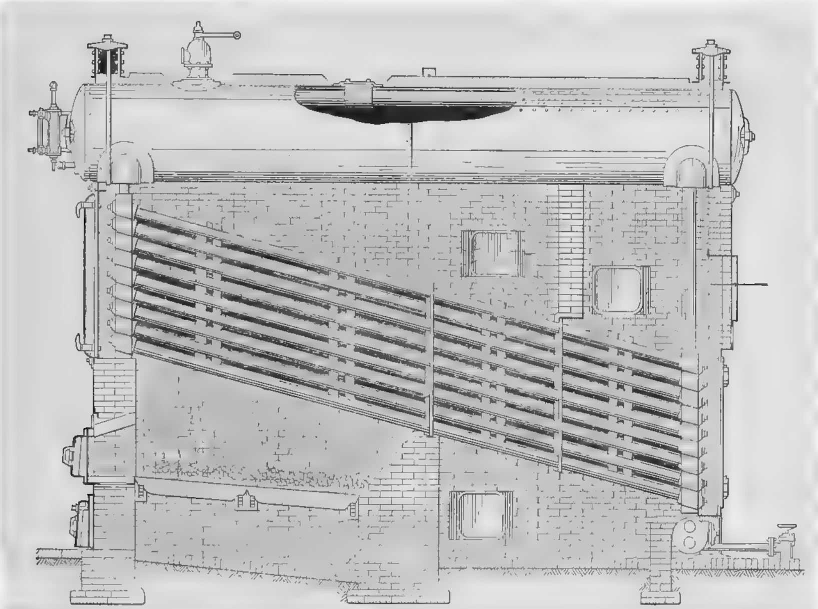


Fig. 48.—Babcock & Wilcox vertical header-boiler.

the steam"; but such surface is always a source of trouble, and is incapable of being graduated to the varying requirements of the steam. No part of a boiler not exposed to water on the one side should be subjected to the heat of the fire upon the other, as the unavoidable unequal expansion necessarily weakens the metal and is a serious source of danger. Hence a boiler which makes dry steam is to be preferred to one that dries steam which has been made wet.

The vertical cylindrical tubular boiler is a convenient type for cramped fire-room space and for portable use. It is most suitable for

small units of power, but is not considered to be of economical form. We illustrate three models which cover their essential differences.

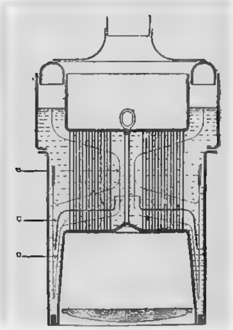


FIG. 49.—Vertical submerged-head boiler.

First, an English design with submerged tubes and enlarged water- and steam-spaces, and in which a diaphragm is inserted among the tubes to divert the circulation across the tubes and clear the tube-head from accumulation of steam.

Secondly, a submerged-tube vertical boiler as ordinarily constructed, in which all the tube-surface and the surface of the furnace divided by 10 equal the boiler horse-power.

Thirdly, the one more commonly in use, the through-tube model, in which the upper ends of the tubes are exposed to undue temperature and to the troubles arising from overheating the upper ends and tube-head, which condition weakens the expanded joints, causing leakage. This, together with the difficulty of cleaning the tubes on the inside and of removing the scale from the outside, or clearing the fire-tube

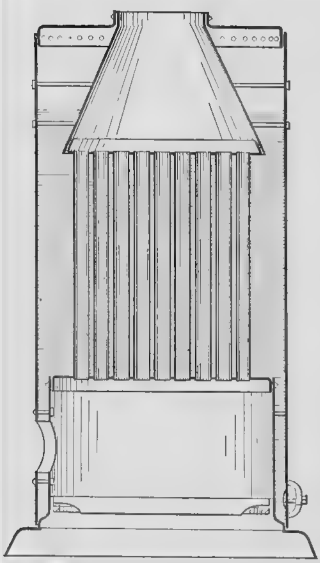


FIG. 50.—Submerged-tube vertical boiler.

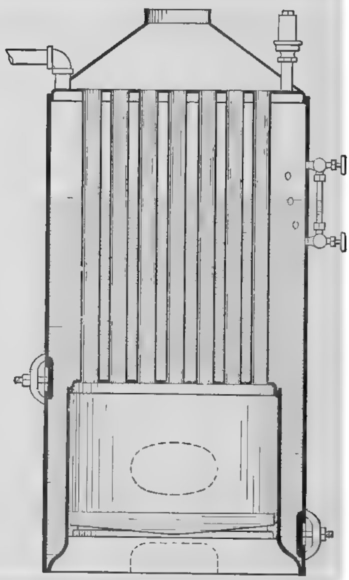


FIG. 51.—Vertical-tube boiler.

sheet from scale and mud that fall upon it, is a serious drawback in the use of this type of boiler, except for temporary or portable necessity.

The water-surface of such boilers is small for the quiet delivery of steam; foaming is waste, and they should not be forced beyond two-thirds of their rated power.

THE HORSE-POWER RATING OF BOILERS

The work of a boiler to convert water into steam requires some unit to represent its general efficiency as a steam-producer. The method of Watt, now abandoned, was the evaporation of 1 cubic foot of water per hour to equal a boiler horse-power. The method of to-day is one that figures on evaporating 30 pounds of water per hour from 100° F. and at 70 pounds gauge-pressure as equivalent to 1 boiler horse-power. This standard is also equivalent to the evaporation of 34.5 pounds of water per hour from and at 212° F. As a boiler can in no way develop power of itself, it would seem that to assign to it the term "horse-power" is illogical, because economy in the use of steam must depend upon the engine alone, and economy varies greatly with the various types of engines.

The term "horse-power" as applied to a boiler seems justified, however, as a matter of convenience, and probably conveys as intelligent an idea as to the power the boiler is able to furnish as any other term.

The power of a boiler to make steam depends upon the amount of heat generated in the furnace, and on the proportion of that heat which is transmitted to the water in the boiler.

The amount of heat liberated through combustion depends upon the quality of the fuel, the rate of combustion, and the size of the grate.

The rate of combustion varies with the draught, quality of coal, and the skill with which the fire is handled. As a general rule a moderate rate of combustion is preferable, as the combustion is more likely to be complete and the heating-surfaces are thus permitted to take up a larger portion of the heat produced, while if the combustion is too rapid a large amount of heat escapes to the stack. On the other hand, when the combustion is too slow a considerable excess of air is admitted to the furnace through the grates and the loss of heat by radiation and conduction is proportionately increased.

The heating-surface of a boiler is a factor which also requires consideration. The heating-surfaces of the various kinds of boilers differ in their efficiency; thus, for instance, the tubes of a return tubular boiler are not equal in radiating value to the shell for equal areas. Neither can both ends of a tube be of equal value, as the value decreases with the length of the tube. It is therefore of little advantage to have the length of the tube more than fifty times its diameter.

The rating of a boiler as now sold is figured from the amount of its heating-surface, allowing from 11 to 12 square feet per horse-power. It is evident that this method of rating is an invitation to boiler-makers to increase the heating-surface at the expense of the boiler capacity.

This has no bearing upon the power that can be obtained from a horse-power of the boiler. The type and model of the engine in its economy of steam used in pounds per horse-power are the real factors that give the value in power that can be obtained from a boiler horse-power; so that a boiler horse-power divided by a steam-engine horse-power in pounds of steam, equals the steam-engine horse-power available per boiler horse-power.

HEATING- AND GRATE-SURFACE FOR BOILERS

The amount of heating-surface per horse-power varies very much in the different types of boilers and with the amount of fuel burned per square foot of grate. The square feet of heating-surface and

TABLE VI.—APPROXIMATE PROPORTION OF HEATING-SURFACE AND GRATE-SURFACE PER HORSE-POWER, ETC., OF VARIOUS TYPES OF BOILERS.

TYPE OF BOILER.	Square feet of heating-surface per horse-power.	Coal per square foot of heating-surface.	Relative economy.	Relative rapidity of steaming.	Heating-surface per square foot of grate.	Pounds of coal per square foot of grate.	Pounds of water per pound of coal.
Water-tube.....	10 to 12	.3	1.00	1.00	35 to 50	12 to 20	9 to 12
Cylind'l tubular.	14 " 16	.25	.91	.60	25 " 35	10 " 15	8 " 11
Vertical tube....	15 " 20	.25	.80	.60	25 " 30	10 " 15	8 " 10
Locomotive.....	12 " 16	.275	.85	.55	50 " 100	20 " 40	8 " 11
Flue	8 " 12	.4	.79	.25	20 " 25	10 " 20	8 " 10
Plain cylindrical.	6 " 10	.5	.69	.20	15 " 20	15 " 25	7 " 9

grate-surface are so variable in the various types and many designs of the same type, that no condition as to actual performance or efficiency of any boiler can be made except as deduced from the actual work of one of its own type and model under equal conditions of operation.

In the foregoing table are values nearly covering the limits of practical work with various types of boilers.

THE INDICATORS OF BOILER-CONTROL

The safety-valve, the pressure-gauge, and the water-gauge are the safety-indicators of all steam-generators and as such are to be watched with all the care of the engineer as indicating what is the condition between the furnace and the engine.

The size of the safety-valve is an important matter, and it is well to consider the area of the grate, the weight of fuel burned, and the steam-pressure when calculating the required area of a safety-valve, because, other things being equal, the volume of steam generated in a given time will depend upon the weight of the coal burned, and the velocity of escape will depend upon the pressure.

A general rule or formula given by Professor Thurston is: Area $= \frac{0.5w}{p+10}$, in which w = weight of steam made per hour in pounds, and p the gauge-pressure. Another formula, of unknown source, is based upon the grate-surface and gauge-pressure: Area of valve $= \frac{22.5G}{p+8.62}$, in which G = grate-surface in square feet; p = gauge-pressure.

When the area of the grate and the steam-pressure are not considered, 1 square inch of valve-area should be provided for each 3 square feet of grate-surface for spring-loaded or "pop" valves, and 1 square inch of valve-area for each 2 square feet of grate-surface for the lever-and-weight variety. A safety-valve should be proportioned for the lowest regular pressure to be carried because steam of higher pressure possesses a smaller volume and escapes at a much higher velocity, so that a smaller valve will discharge the same weight of steam in less time; therefore, as the pressure becomes higher the valve may be made smaller.

In the following table are given the safety-valve areas in square inches per square foot of grate and various pressures based on the velocity and weight of issuing steam at the different pressures.

TABLE VII.—AREAS OF LEVER SAFETY-VALVES FOR EACH SQUARE FOOT OF GRATE-SURFACE.

Gauge-pressure.	Area in square inches.	Gauge-pressure.	Area in square inches.	Gauge-pressure.	Area in square inches.	Gauge-pressure.	Area in square inches.
15	1.250	65	.468	115	.288	165	.208
20	1.071	70	.441	120	.277	170	.202
25	.937	75	.416	125	.267	175	.197
30	.833	80	.394	130	.258	180	.192
35	.750	85	.375	135	.250	185	.187
40	.681	90	.357	140	.241	190	.182
45	.625	95	.340	145	.234	195	.178
50	.576	100	.326	150	.227	200	.174
55	.535	105	.312	155	.220	225	.166
60	.500	110	.300	160	.214	250	.158

In Fig. 52 is shown the lever safety-valve, the lever of which is of the third order. O is the fulcrum; A the distance from the fulcrum to the centre of the valve; B the distance from the fulcrum to the centre of gravity of the lever; C the distance from the fulcrum to the centre of the weight; D total length of the lever, and S the diameter of the valve-opening, all in inches. G = the weight of the lever at its centre of gravity, and W the weight of ball; V = the weight of the valve and spindle; P = pressure in pounds per square inch.

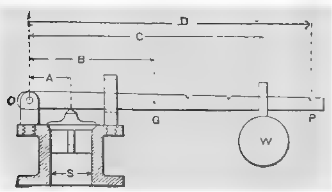


FIG. 52.—Lever safety-valve.

$$W=\frac{S^2\times .7854\times P\times A-(G\times B)-(V\times A)}{C}$$

$$C=\frac{S^2\times .7854\times P\times A-(G\times B)-(V\times A)}{W}$$

In Fig. 53 is shown a differential safety-valve in which the enlarged area of the upper valve compensates for the differential tension of the spring upon opening the valve, thus causing the valve to open wide without increase of boiler-pressure.

Another form of spring safety-valve, known as the "pop" or reactionary valve, of which the Ashcroft is a good example, is one in

which the steam issuing from under the valve is deflected by a curved lip or flange in such a manner as to cause an additional pressure by its reaction that aids effectively in raising the valve. The pressure at which a "pop" valve will blow cannot, as a rule, be as closely

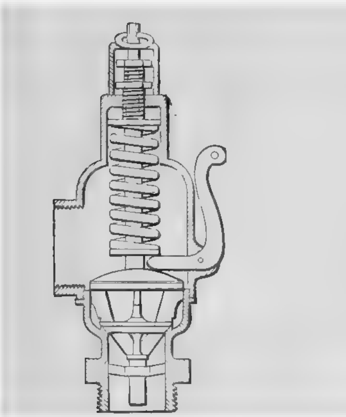


FIG. 53.—Differential safety-valve.

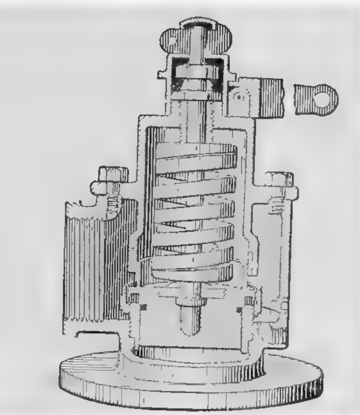


FIG. 54.—"Pop" safety-valve.

estimated as with the lever-and-weight style, and must therefore be finally adjusted by trial. Fig. 54 shows a section of the consolidated pop safety-valve with wing-guides. The seat is narrow and at an angle of 45°, above which are the enlarged lipped area and shield.

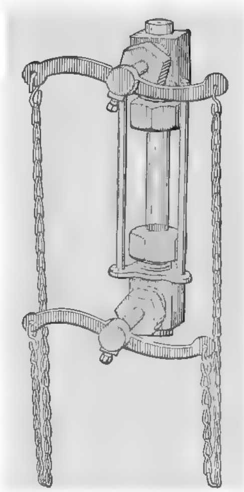


FIG. 55. — Quick-closing water-gauge.

Water-gauges and their position, with the facilities for keeping them in perfect condition, are essential to the welfare of a steam-plant. Their length should correspond with and cover the range of water-level assigned for the different sizes and types of boilers. In all cases they should be fixed to water-columns or stand-pipes containing the gauge-cocks; although with the ordinary vertical boilers this is not always a fast rule. Their pipe-connections should be so arranged that dry steam enters the top of the water-column and water enters the bottom from a quiet part of the boiler-water, with blow-off valves for both water-column and water-gauge.

A water-column should be blown out at least once a day, and as often as three or four times a day, depending upon the quality of the feed-water employed. The gauge-cocks should be opened after blowing out the column or the glass to see that the water-level in the glass tallies with the level indicated by the gauge-cocks.

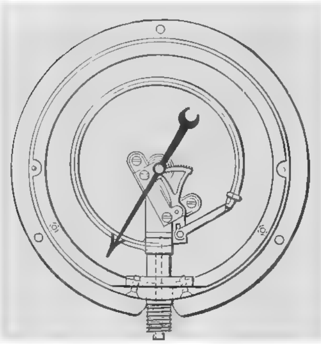


Fig. 56.—Bourdon gauge.

Pressure-gauges should as a rule be placed convenient for observation, with the shortest piping possible, and with a siphon beneath the gauge for its protection from injury from steam within its spring. A cock on the gauge is necessary, and if the pipe is of a length to accumulate water, a pet-cock at its lowest point near the gauge serves to blow out any sediment and prove the proper connection of the gauge with the boiler.

One of the most satisfactory and convenient instruments for the engine-room or office, to show the range of boiler-pressure during the daily run, is an Edson recording steam-gauge, which we illustrate in Fig. 57 and detail in Fig. 58. The diaphragm D is so corru-

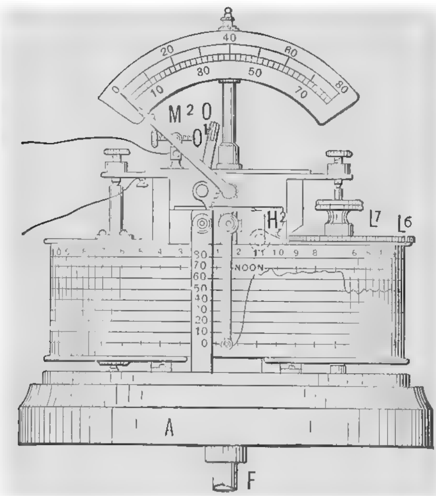


Fig. 57.—Edson recording gauge.

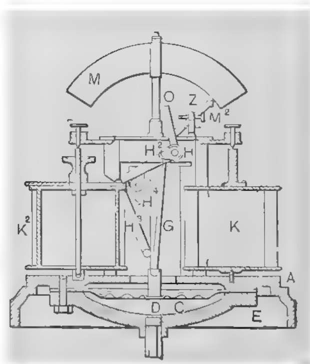


Fig. 58.—Section of gauge.

gated that its movement under pressure shall be practically uniform for equal increases of pressure. From this diaphragm a connecting-rod, G,

actuates a small crank, H, the shaft of which bears an open segment, which actuates a pinion on the arbor of an index showing the pressure on the diaphragm. At the same time, by means of levers H^2, H^3, vertical movement is communicated to a pencil-point, which records gradually on a graduated paper ribbon the pressure shown by the index as being on the diaphragm. The paper strip has given to it by clock-work a regular motion from the drum K to K^2, and has marked on it vertical spaces corresponding to hours. By this means, not only the index-hand shows the pressure put on the gauge, but the pencil makes a continuous record showing all fluctuations and when they occurred. There is also an electrical-alarm attachment by means of which, when the pressure passes a certain limit, a bell is rung.

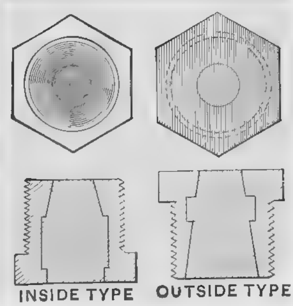


FIG. 59.—Safety-plugs.

Fusible plugs are in use and are required by law to be applied to boilers of sea-going vessels. They are generally composed of pure Banca tin, which melts at 443° F., incased in a brass shield and screwed into the boiler-sheet from the water side and at the highest point subject to exposure to the furnace-heat by low water. In Fig. 59 is shown an inside and an outside fusible plug of the model required by the United States Inspection Service. No composition metal is allowed; the diameter of the water-end of the tin plug is about one-half inch.

The feed-pipe should enter a boiler at a convenient point and be so arranged on the inside as to prevent as much as possible the immediate contact of the feed-water with the highly heated furnace-plates or hot part of the tubes. There are many designs of the arrangement of feed- and blow-off pipes, adapted to the different types and models of boilers in use, so that no fast rule can be quoted.

STRENGTH OF CYLINDRICAL SHELL-BOILERS

As there are apprehensions among some engineers in regard to the direction of the strains in the shell of a cylindrical boiler, we illustrate in Fig. 60 these conditions, which are shown by the directions

of the arrows in the upper and lower half of the shell. It will be seen that the actual direction of the pressure is radial; but the resultants of the directions of the arrows are only fully effective at the points of their angles of position and only as the sine of the angle for any single collective point in the circumference, and for the diameter, as in the diagram; the results due to the sines of the radial stresses in the upper half are shown in the direction of the arrows in the lower half of the diagram. The pressure in pounds per square inch, multiplied by the semidiameter of the shell in inches, equals the strain in a section of the shell 1 inch wide.

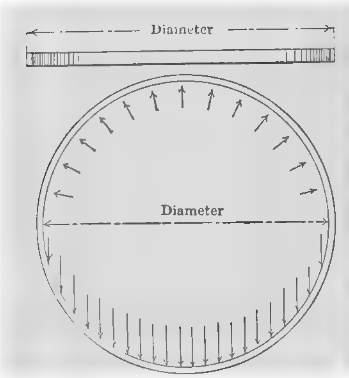


FIG. 60.—Strains in a boiler-shell.

The resistance in the shell is the tensile strength per square inch, multiplied by the thickness in decimals of an inch for the sheet alone and an allowance made for the loss in strength by the riveted seam.

For single-riveted seams an allowance of from 40 to 45 per cent. of the tensile strength of the sheet should be made and for double-

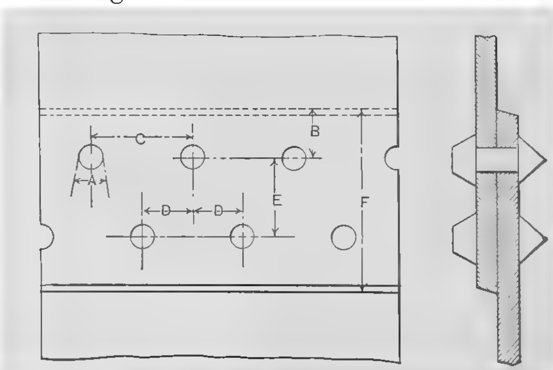


FIG. 61.—Double-lap joint.

riveted seams an allowance of 33 per cent.; making their tensile strength 55 and 67 per cent. of that of the plate. The strength of the single-riveted girt-seam is much greater for any size shell than a single- or double-riveted longitudinal seam; besides, the head-support from the tubes is fully equal to the strain within their area. The strength of

the longitudinal seam in all forms of boiler-shells, steam-drums, riveted pipes and tanks, determines to a marked degree the pressure which the structure is capable of carrying continuously and with safety. Boilers are now so generally made of steel plates and rivets that we give some details of the make-up of these seams. In Fig. 61 is

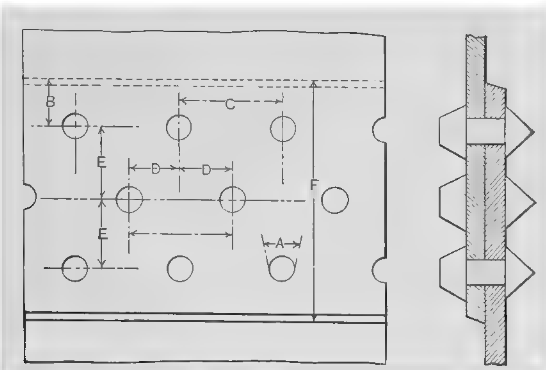


Fig. 62.—Triple-lap joint.

shown the lay-out of a double-riveted lap-joint with the rows staggered, which is the strongest joint with two rows of rivets; and in Fig. 62 the lay-out of a triple-riveted seam, with an accompanying table of thicknesses of plates, sizes of rivets and holes, and their distance apart; also the percentage of the strength of the seam in proportion to the strength of the plate.

Table VIII.—Proportions for Boiler-Joints; Double-Riveted.

Thickness of plate.	Diameter.		Centre of hole to edge of plate.	Pitch of Rivets.		Lap of Plates.	Percentage of Joint.	
				Centre to centre zigzag riveting.		Zigzag riveting.	Steel plate.	
	Rivet.	Hole.		Horizontal.	Vertical.		Iron rivets.	Steel rivets.
	A	A		C	E	F		
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	1	$2\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{1}{2}$	72.48	72.48
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$1\frac{1}{8}$	$2\frac{5}{8}$	$1\frac{5}{16}$	$2\frac{5}{8}$	67.01	71.46
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$1\frac{3}{16}$	$2\frac{3}{4}$	$1\frac{3}{8}$	$2\frac{3}{4}$	62.54	70.42
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$1\frac{5}{16}$	$2\frac{7}{8}$	$1\frac{7}{16}$	$2\frac{7}{8}$	59.44	69.55
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{3}{8}$	$2\frac{15}{16}$	$1\frac{15}{32}$	$2\frac{15}{16}$	58.44	68.07
$\frac{9}{16}$	$\frac{15}{16}$	1	$1\frac{1}{2}$	3	$1\frac{1}{2}$	3	57.87	66.65
$\frac{5}{8}$	1	$1\frac{1}{16}$	$1\frac{9}{16}$	$3\frac{1}{8}$	$1\frac{9}{16}$	$3\frac{1}{8}$	56.46	66.00
$\frac{11}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{11}{16}$	$3\frac{5}{16}$	$1\frac{21}{32}$	$3\frac{5}{16}$	54.29	66.05
$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{3}{4}$	$3\frac{3}{8}$	$1\frac{11}{16}$	$3\frac{3}{8}$	54.42	64.82

PROPORTIONS FOR BOILER-JOINTS; TRIPLE-RIVETED.

$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	1	$3\frac{1}{2}$	$1\frac{1}{2}$	5	79.14	80.34
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$1\frac{1}{8}$	$3\frac{5}{8}$	$1\frac{9}{16}$	$5\frac{3}{8}$	72.74	79.25
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$1\frac{3}{16}$	$3\frac{3}{4}$	$1\frac{5}{8}$	$5\frac{5}{8}$	68.79	78.37
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$1\frac{5}{16}$	$3\frac{7}{8}$	$1\frac{11}{16}$	6	66.18	77.46
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{3}{8}$	4	$1\frac{13}{16}$	$6\frac{3}{8}$	64.39	76.56
$\frac{9}{16}$	$\frac{15}{16}$	1	$1\frac{1}{2}$	$4\frac{1}{8}$	$1\frac{7}{8}$	$6\frac{3}{4}$	63.15	75.78
$\frac{5}{8}$	1	$1\frac{1}{16}$	$1\frac{9}{16}$	$4\frac{1}{4}$	2	$7\frac{1}{8}$	62.27	75.04
$\frac{11}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{11}{16}$	$4\frac{3}{8}$	$2\frac{1}{16}$	$7\frac{1}{2}$	61.64	74.27
$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{3}{4}$	$4\frac{1}{2}$	$2\frac{1}{8}$	$7\frac{3}{4}$	61.22	73.63

The hydraulic test of a boiler should be at a pressure not more than one and a half times the working pressure.

TABLE IX.—SAFE WORKING PRESSURE FOR WELL-MADE CYLINDRICAL TUBULAR BOILERS WITH STEEL SHELLS; DOUBLE- AND TRIPLE-RIVETED.

Diameter.	Thickness.	Steel shell. Iron rivets.		Steel shell. Steel rivets.	
36	$\frac{1}{4}$	111	121	111	123
	$\frac{5}{16}$	128	139	137	151
38	$\frac{1}{4}$	105	115	105	116
	$\frac{5}{16}$	121	132	129	144
40	$\frac{1}{4}$	100	109	100	110
	$\frac{5}{16}$	115	125	123	136
42	$\frac{1}{4}$	95	104	95	105
	$\frac{5}{16}$	110	119	117	130
44	$\frac{1}{4}$	91	99	91	100
	$\frac{5}{16}$	105	114	112	124
46	$\frac{1}{4}$	87	95	87	96
	$\frac{5}{16}$	100	109	107	119
48	$\frac{5}{16}$	96	104	102	114
	$\frac{3}{8}$	107	118	121	135
50	$\frac{5}{16}$	92	100	98	109
	$\frac{3}{8}$	103	113	116	129
52	$\frac{5}{16}$	89	96	95	105
	$\frac{3}{8}$	99	109	112	124
54	$\frac{5}{16}$	85	93	91	101
	$\frac{3}{8}$	96	105	108	120

Diameter.	Thickness.	Steel shell. Iron rivets.		Steel shell. Steel rivets.	
56	$\frac{5}{16}$	82	89	88	97
	$\frac{3}{8}$	92	101	104	116
58	$\frac{5}{16}$	79	86	85	94
	$\frac{3}{8}$	89	98	100	112
60	$\frac{5}{16}$	77	83	82	91
	$\frac{3}{8}$	85	95	97	108
62	$\frac{3}{8}$	83	92	94	104
	$\frac{7}{16}$	92	103	108	120
64	$\frac{3}{8}$	81	89	91	101
	$\frac{7}{16}$	89	100	105	117
66	$\frac{3}{8}$	78	86	88	98
	$\frac{7}{16}$	87	97	102	113
68	$\frac{3}{8}$	76	84	86	95
	$\frac{7}{16}$	80	94	99	110
70	$\frac{3}{8}$	74	81	83	92
	$\frac{7}{16}$	82	91	96	107
72	$\frac{3}{8}$	72	79	81	90
	$\frac{7}{16}$	79	89	93	104
	$\frac{1}{2}$	89	98	104	117

Perhaps the most important detail in boiler-construction is the bracing of flat surfaces. The end-surfaces of steam-boilers are stayed by means of braces extending from the heads to the shell or by longitudinal stays extending from head to head. In all flue-boilers in which the flues are riveted to the heads, the flues themselves act as stays, and usually have strength enough to dispense with other stays below the water-line, except in very large boilers or those adapted to very high pressures. The holding-power of wrought-iron tubes expanded in the heads is sufficient to withstand any working pressure

occurring in that portion of the boiler in which the tubes are located. This refers especially to stationary boilers.

Flanging the edges of a boiler-head increases its stiffness along the outer edge, and for this reason 2 inches of this outer flanged

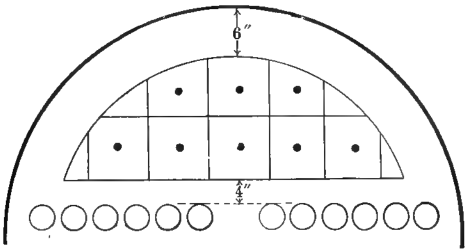


FIG. 63.—Stayed boiler-head.

surface may be left to take care of itself. The influence of the flange extends inward, and no braces need be located within 4 inches of the flange radius for pressures less than 150 pounds per square inch.

The holding-power of the tubes imparts sufficient stiffness to the boiler-head not to require braces nearer than 4 inches, so that in all ordinary calculations the area to be supported would be represented

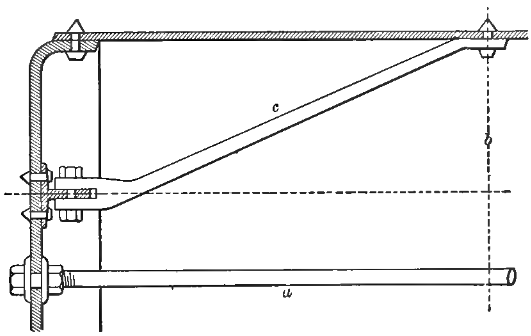


FIG. 64.—Diagonal and direct stay.

by the segment of a circle (as shown in Fig. 63) of 6 inches less radius than the boiler-head and its base-line or chord 4 inches above the tubes.

The location of the stay-centres is not easily worked out except on the drawing-board, but the area to be stayed in any given case can

be obtained by computation of the area within the lines as above described. For example, for a boiler-head 58 inches in diameter with a required braced area of 353.77 square inches, the load to be carried by the braces will be equal to the area found multiplied by the maximum safe working pressure in pounds. If the pressure is to be, say, 130 pounds per square inch, then the total load will be $353.77 \times 130 = 45{,}990$ pounds. No boiler-brace should be allowed a greater stress than 6,000 pounds per square inch, measured at the smallest part. The number of braces required may be found by dividing the total load by what one brace will safely carry, which in this case is $45{,}990 \div 6{,}000 = 7.65$, say 8 braces; that is, it will require 8 braces having an area of 1 square inch, which corresponds to about $1\frac{1}{8}$ inches diameter.

The surface or area supported by each brace is found by dividing the whole area to be supported by the number of braces, which gives

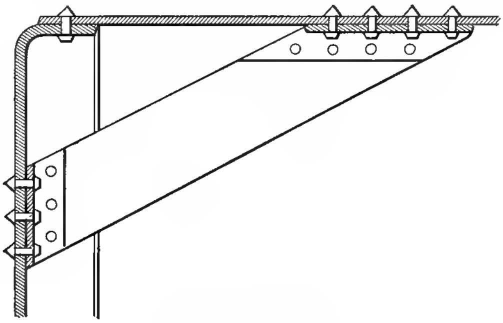


FIG. 65.—Gusset-brace.

$353.77 \div 8 = 44.22$ square inches. The square root of this number will give the distance between the braces or the pitch, which is 6.64 inches, or $6\frac{11}{16}$ inches.

Table X gives the proper distance between the stays in a boiler for different maximum pressures.

The table gives the size of stays corresponding to the several pressures when the stays run at right angles to the head, as at *a* in Fig. 64; but when they are placed at an angle, as at *c*, their holding-power for a given area of cross-section is considerably reduced, and in order to maintain the holding-power the area of the stays must

be increased. The required area of a diagonal stay may be obtained by dividing the area of the direct one by the cosine of the angle that the brace bears to the axis of the shell.

TABLE X.—DIRECT STAYS FOR BOILERS.

Diameter in inches.	Area, square inches.	Working strength.	Number of inches square each brace will sustain under the following pressures:			
			75 lbs.	100 lbs.	125 lbs.	150 lbs.
$\frac{7}{8}$	.60	3,600 lbs.	7.	6.	5.4	4.9
1	.78	4,712 "	7.9	6.9	6.1	5.6
$1\frac{1}{8}$	.99	5,964 "	8.9	7.7	6.9	6.4
$1\frac{1}{4}$	1.23	7,362 "	9.9	8.6	7.7	7.0
$1\frac{3}{8}$	1.48	8,880 "	10.7	9.5	8.5	7.7
$1\frac{1}{2}$	1.77	10.620 "	11.9	10.4	9.2	8.5

The stresses in boilers and the constructive details, as to the strength of shells, seams, heads and braces, are given here as belonging to the special duties and practice of the engineer, as an inspector of the steam plant in his charge.

There is much more that might be written in regard to the details of boiler construction, which, for the information of engineers interested, we refer to works treating exclusively upon this subject.

CHAPTER V

BOILER-CHIMNEY AND ITS WORK

THE power of a chimney to create draught depends somewhat on its form as well as height; but the main force of draught is in the difference of outside and inside temperatures. The ratio of wall-surface to its area is in evidence in the draught problem; and in general terms a round chimney is first in efficiency because its wall-surface is least in proportion to its area, the ratio being for equal areas about 13 per cent. greater wall-surface in a square chimney.

Theoretically the strongest draught in a well-proportioned chimney is claimed to be obtained by a difference of absolute temperatures of 25 to 12, or when the atmospheric temperature is 60° and the chimney temperature 622° F. For internal chimney temperatures above 622° the densities of the gases decrease faster than the velocity increases, so that the weight of the gases passing up the chimney is at a maximum at about this temperature; but the draught-pressure increases with the height within reasonable limits.

The effective area of a chimney varies inversely as the square root of the height, and is less than the actual area, owing to the friction of the gases against the walls, on the basis that this is equal to a layer of gas 2 inches thick on the wall-surface. The formula: Effectual area $= \frac{0.3\ H}{\sqrt{h}}$ also equals $A - 0.6\sqrt{A}$, in which H is the boiler horse-power, h height of chimney, and A the actual area. Also, the boiler horse-power of a chimney may be computed from the formula: $H = 3.33E\sqrt{h}$, and the height for any horse-power from $h = \left(\frac{0.3\ H}{E}\right)^2$, in which E is the effective area.

The diagram, Fig. 66, shows the draught in inches of water for a chimney 100 feet high with different temperatures above the external atmosphere, say 60° F. The vertical lines represent the chimney temperatures above 60° F. and the horizontal lines are 20 to 1

inch. The upper curved line shows the ratio of the flow of gases at the various temperatures along the vertical lines in pounds per hour, which may be computed by multiplying the height of the curve at any given chimney temperature above that of the atmosphere by

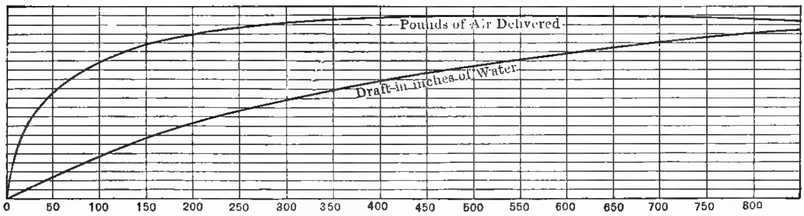


FIG. 66.—Draught and weight of chimney-gases.

the vertical scale in decimals of an inch, and by 1,000 times the effective area in square feet, and by the square root of the height in feet. $S\ 1{,}000E\sqrt{h}$ = pounds per hour, in which S = decimals of an inch on the vertical scale, E = effective area, and h = height of chimney. The pressure-curve in decimals of an inch of water is computed from the formula: $h\left(\frac{7.6}{t_a}-\frac{7.9}{t_c}\right)$, in which h = height of chimney in feet, t_a is the absolute temperature of the air entering the furnace, and t_c the absolute temperature of the chimney-gases.

For example, for a chimney 100 feet high with air and gas temperatures of 60° and 660° F., $100\left(\frac{7.6}{520^\circ}-\frac{7.9}{1{,}120^\circ}\right)=.756$ of an inch water-pressure.

From this formula Table XI has been computed for external temperatures of from 10° to 90° F., and for chimney temperatures from 240° to 700° F.

For any other height of chimney than 100 feet, the water-pressure will be approximately in proportion to the height, so that the pressures in the table columns at the junction of the external and chimney temperatures, multiplied by the decimal representing the proportion to 100 feet, will give the required water-pressure.

For example, for the respective temperatures of 60° and 600° F., and 160 feet in height, $1.6\times.716=1.14$ inches water-pressure.

TABLE XI.—DRAUGHT-PRESSURE IN INCHES OF WATER.
IN A CHIMNEY 100 FEET HIGH.

Temperature in chimney.	Temperature of external air. (Barometer, 30 inches.)								
	10°	20°	30°	40°	50°	60°	70°	80°	90°
240°	.488	.451	.421	.388	.359	.330	.301	.276	.250
260	.528	.484	.453	.420	.392	.363	.334	.309	.282
280	.549	.515	.482	.451	.422	.394	.365	.340	.313
300	.576	.541	.511	.478	.449	.420	.392	.367	.340
320	.603	.568	.538	.505	.476	.447	.419	.394	.367
340	.638	.593	.569	.530	.501	.472	.443	.419	.392
360	.653	.618	.588	.555	.526	.497	.468	.444	.417
380	.676	.641	.611	.578	.549	.520	.492	.467	.440
400	.697	.662	.632	.598	.570	.541	.513	.488	.461
420	.718	.684	.653	.620	.591	.563	.534	.509	.482
440	.739	.705	.674	.641	.612	.584	.555	.530	.503
460	.758	.724	.694	.660	.632	.603	.574	.549	.522
480	.776	.741	.710	.678	.649	.620	.591	.566	.540
500	.791	.760	.730	.697	.669	.639	.610	.586	.559
550	.835	.801	.769	.738	.698	.679	.652	.625	.600
600	.872	.838	.806	.775	.735	.716	.689	.662	.637
650	.906	.872	.840	.809	.769	.750	.723	.696	.671
700	.936	.902	.870	.839	.799	.780	.753	.726	.701

A simple form of draught-gauge is shown in Fig. 67. It consists of a small glass tube bent into a U shape, one-half filled with water, with a scale of tenths of an inch fixed between the legs; or the actual difference in level of the water may be measured when one of the legs is connected to the chimney by a tube. Usually a piece of ¼-inch iron pipe is passed through a hole in the main flue and connected to the gauge with a piece of rubber tube.

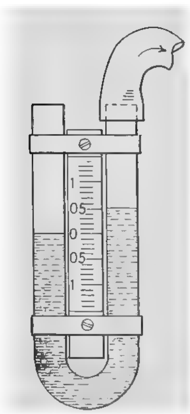


FIG. 67.
Draught-gauge.

The size and height of a chimney and its boiler horse-power depend upon the amount of coal assumed to be burned per horse-power, which requires a variable size and height to meet the assumed economy of a steam-plant.

Table XII is based on the average consumption of 5 pounds of coal per hour per horse-power, which is assumed to be the maximum amount in any well-proportioned power-plant. For any less amounts of coal burned, a reduction in chimney height and area or an increase in the boiler-

TABLE XII.—SIZE AND HEIGHT OF CHIMNEYS FOR THE HORSE-POWER RATING OF BOILERS.

Diameter, inches.	Area, square feet.	Effective area, square feet.	Height of Chimney in Feet.													Equivalent side of square chimney, inches.
			50	60	70	80	90	100	110	125	150	175	200	225	250	
			For the commercial horse-power of boilers.													
18	1.77	.97	23	25	27	29	...	...	....	....	....	....	....	....	....	16
21	2.41	1.47	35	38	41	44	...	...	....	....	....	....	....	....	....	19
24	3.14	2.08	49	54	58	62	66	...	....	....	....	....	....	....	....	22
27	3.98	2.78	65	72	78	83	68	...	....	....	....	....	....	....	....	24
30	4.91	3.58	84	92	100	107	113	119	....	....	....	....	....	....	....	27
33	5.94	4.48	..	115	125	133	141	149	156	....	....	....	....	....	....	30
36	7.07	5.47	..	141	152	163	173	182	191	204	....	....	....	....	....	32
39	8.30	6.57	..	...	183	196	208	219	229	245	....	....	....	....	....	35
42	9.62	7.76	..	...	216	231	245	258	271	289	316	....	....	....	....	38
48	12.57	10.44	..	...	...	311	330	348	365	389	426	....	....	....	....	43
54	15.90	13.51	..	...	...	...	427	449	472	503	551	595	....	....	....	48
60	19.64	16.98	..	...	...	...	536	465	593	632	692	748	....	....	....	54
66	23.76	20.83	..	...	...	...	...	694	728	776	849	918	981	....	....	59
72	28.27	25.08	..	...	...	...	...	835	876	934	1,023	1,105	1,181	1,253	....	64
78	33.18	29.73	..	...	...	...	...	...	1,038	1,107	1,212	1,310	1,400	1,485	1,565	70
84	38.48	34.76	..	...	...	...	...	...	1,214	1,294	1,418	1,531	1,637	1,736	1,830	75
90	44.18	40.19	..	...	...	...	...	...	....	1,496	1,639	1,770	1,893	2,008	2,116	..
96	50.27	46.01	..	...	...	...	...	...	....	1,712	1,876	2,027	2,167	2,298	2,423	..
102	56.75	52.23	..	...	...	...	...	...	....	1,944	2,130	2,300	2,459	2,609	2,750	..
108	63.62	58.83	..	...	...	...	...	...	....	2,090	2,399	2,592	2,771	2,939	3,098	..
114	70.88	65.83	..	...	...	...	...	...	....	....	2,685	2,900	3,100	3,288	3,466	..
120	78.54	73.22	..	...	...	...	...	...	....	....	2,986	3,226	3,448	3,657	3,855	..
132	95.03	89.18	..	...	...	...	...	...	....	....	3,637	3,929	4,200	4,455	4,696	..
144	113.10	106.72	..	...	...	...	...	...	....	....	4,352	4,701	5,026	5,331	5,618	..

horse-power columns may be made, proportionable to the amount of coal assigned per horse-power hour.

For example, for a power-plant assumed to use but $2\frac{1}{2}$ pounds of coal per horse-power hour, any boiler-power in the table may be divided by 2, and its amount compared with other chimney sizes and heights for selecting the required size and height of chimney.

The formula by which the table has been computed is: Boiler horse-power $= (3.33A - 0.6\sqrt{A})\sqrt{h}$, or $3.33E\sqrt{h}$; $E = A - 0.6\sqrt{A}$; $D = 13.54\sqrt{E} + 4$; $h = \left(\frac{0.3H}{E}\right)^2$; $S = 12\sqrt{E} + 4$, in which A = area of chimney in square feet; D = diameter in inches; E = effective area; S = side of square chimney of equal effective area; h = height in feet; H = horse-power.

Fig. 68 is an example of a steel chimney about 200 feet high, lined with brick, and anchored to a deep foundation by 12 long bolts $1\frac{1}{2}$ inches in diameter. It is 13 feet diameter inside and equal to a draught for 5,000 horse-power.

Fig. 69 is an example of a brick chimney of varying dimensions suitable for a power-plant, as shown; it is subject to details suitable for any required power.

The main point to be observed in the construction of a chimney, after the height and internal diameter are fixed, is its stability or power to resist with safety the overturning force of the highest winds, which requires a proportionate relation between the weight, height, breadth of base, and exposed area of the chimney. This relation is expressed in the equation $C\frac{d\,h^2}{b} = W$, in which d = the average breadth of the shaft; h = its height; b = the breadth of base—all in feet; W = weight of chimney in pounds, and C = a coefficient of wind-pressure per square foot of area. This varies with the cross-section of the chimney, and = 56 for a square, 35 for an octagon, and 28 for a round chimney. Thus, a square chimney of average breadth of 8 feet, 10 feet wide at base and 100 feet high, would require to weigh $56 \times 8 \times 100 \times 10 = 448{,}000$ pounds to withstand any gale likely to be experienced. Brickwork weighs from 100 to 130 pounds per cubic foot; hence such a chimney must average 13 inches thick to be safe. A round stack could weigh half as much, or have less base.

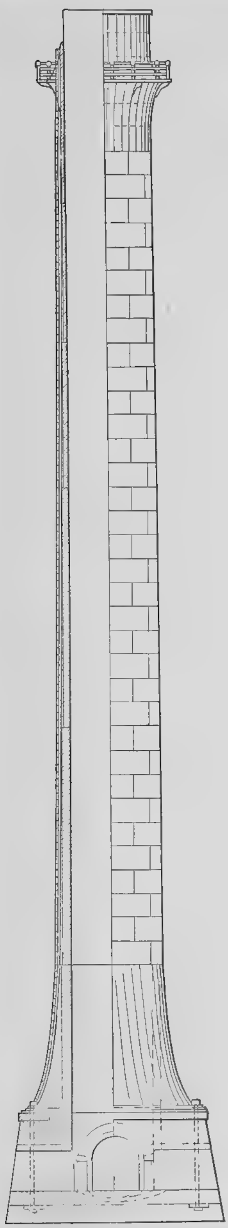


FIG. 68.—Steel chimney.

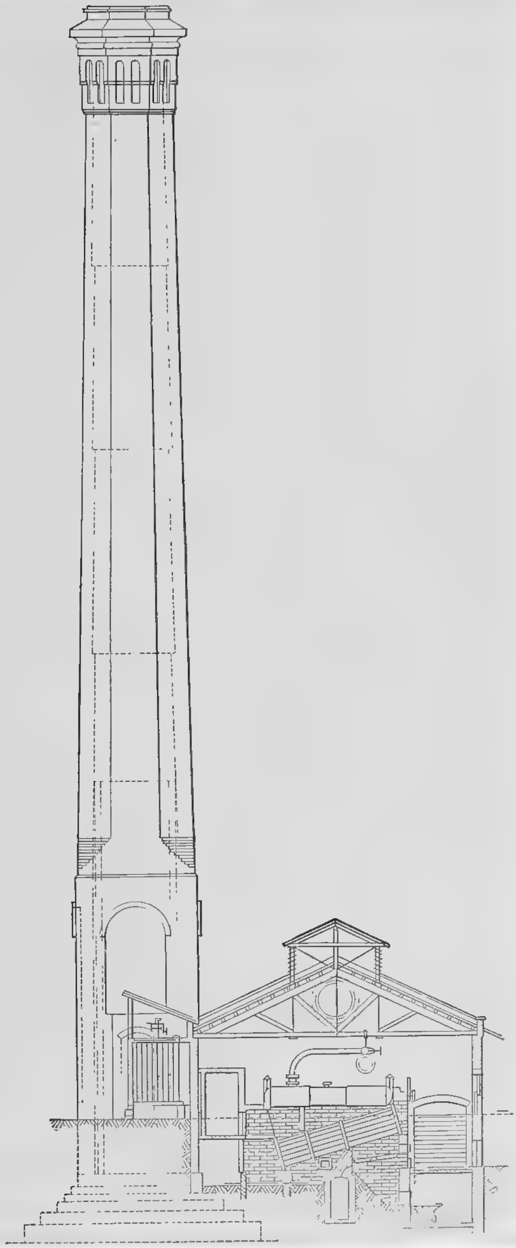


FIG. 69.—Brick chimney and power-plant.

The external diameter of a brick chimney at the base should be one-tenth the height, unless it be supported by some other structure. The "batter" or taper of a chimney should be from $\frac{1}{16}$ to $\frac{1}{4}$ inch to the foot on each side.

FIRING AND THE CHIMNEY-DRAUGHT

The chimney-draught is one of the first things to be studied in the design of a power-plant, since upon it primarily depend the power and performance of the boilers. The amount of fuel that can be burned in a given time on a square foot of grate-surface depends on the strength of the draught, and draught, coal consumption, and efficiency are so closely allied that a discussion of one naturally brings in the others. If the draught is poor the fires have to be carried thin, and in this way a larger amount of air than is necessary for the combustion of the coal comes up through the grates; this excess air is not only useless, but it entails a loss by lowering the average temperature of the furnace, and the heat will not transfer to the water so rapidly. By careful nursing a heavy fire may be built up with a poor draught, but the average fireman is not a good nurse; he will either poke the fire or leave it alone, and his ideas of when to do these necessary parts of firing are generally as nearly right as are the ideas of many of those who try to show him how. Natural draught gives its best economy with a low rate of combustion, and as a general rule the fires should be carried as thick as the draught will burn without the fires having to be broken up more than once an hour. The stronger the draught, the more coal can be burned before the economical limit is reached.

When forcing the fires with a strong chimney-draught there is the same loss as with thin fires and poor draught, namely, too much air per pound of coal burned, and another and greater loss, that of the increased temperature of the flue-gases to maintain the stronger draught; for after the temperature of the flue-gases exceeds 600° F. it takes more heat to strengthen the draught. Under the conditions found in the ordinary plant with natural draught, the coal burned per square foot of grate-surface per hour averages from 10 to 25 pounds, and the air used per pound of coal runs from 20 to 30 pounds. Twelve pounds of air per pound of coal is all that is

required for its complete combustion, and the average plant uses nearly twice this amount. Anything that will cut down the amount of air used per pound of coal will effect a saving. The most that can be saved in this way is about 5 per cent. of the coal.

With natural draught the only way of limiting the amount of air used is by keeping thick fires. But thick fires are apt to be neglected, because the fireman knows that if the steam gets down he can stir up the fires and have it right up again, while he would manage thin fires in a better manner, because he cannot take any chances with them. Furthermore, there are some kinds of coal which lie on the grates like sand, and it is impossible to get enough draught with a chimney to burn a thick fire of such coal.

This brings us to a consideration of some of the kinds of forced-draught apparatus. When speaking of forced draught the general idea is that the air supplied by a fan or other apparatus will cause a higher rate of combustion of the coal than is possible with the rarefaction of air in a chimney. This was the practice at first, but forced or induced draught is now used for all rates of combustion. When coal is dear more attention is paid to ways of burning cheap coal economically, for some of these coals will evaporate nearly as much water as high-priced coal, and cost less than half as much. The cheapest apparatus for this purpose is some form of steam-jet, either one that produces a partial vacuum in the chimney, as in a locomotive, or one that blows into a closed ash-pit and carries a large body of air in with the steam. But on account of the large amount of steam required for operation, a steam-jet is only advisable where fuel is very cheap.

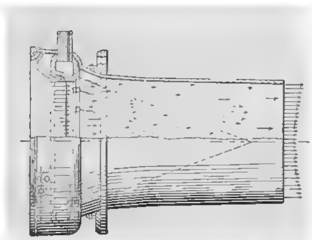


FIG. 70.—Annular steam-blower.

In Fig. 70 is illustrated one of many models of steam-jet blowers, with an annular cast-iron chamber perforated for steam-jets at an angle that projects the jets in a converging direction that draws in the air with a force corresponding with the pressure of the steam. This class of blowers are much in use and are connected with the ash-pit and in short chimneys at their base.

In Fig. 71 is illustrated one of the Korting type of steam-blowers, with a double-nozle air-inlet and double-cone nozle for steam. A needle-valve regulates the flow of steam from the central jet, which

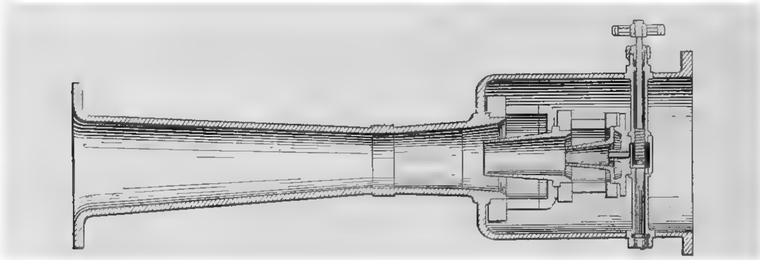


FIG. 71.—Korting steam-blower.

is reenforced by the combined steam and air in the larger nozle, by which a larger volume of air is induced and expanded in the diverging-nozle.

In Fig. 72 is illustrated a low-speed fan-blower of large volume and force. Its particular feature is in the narrow curved blades set in the periphery of the wheel and close together, which prevent local eddies and greatly increase the efficiency of the fan.

A well-designed forced-draught fan will run with less than 1 per cent. of the steam supplied to the engines, while the amount of heat

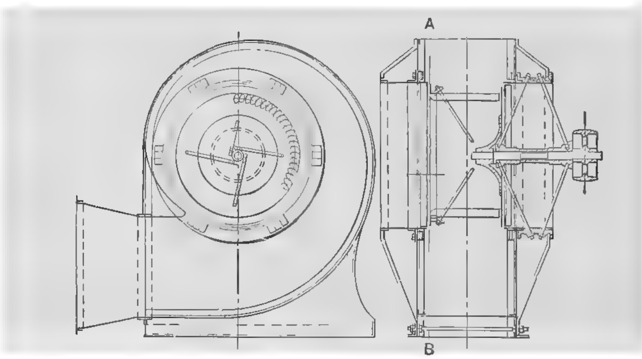


FIG. 72.—Sirocco fan-blower.

that a chimney requires for its operation will be equal to 30 per cent. of the steam supplied to the engines. Practically it is impossible to save this 30 per cent., on account of the cost of apparatus for reducing the temperature of the flue-gases to that of the air enter-

ing the ash-pit. The first saving a fan-draught will make is in the lessened amount of air per pound of coal. A number of tests have been reported where the air supplied was less than 15 pounds to each pound of coal. The temperature of the furnace with a lessened air-supply will be higher; and the heat will be more quickly transferred from the gases to the water of the boiler. The gases will travel more slowly over the heating-surface, and the temperature of the chimney-gases will be lower. Economizers placed between the boilers and the chimney will heat the feed-water and save more than enough to pay for themselves in a short time.

The use of hot air for the furnace effects a saving in fuel, and with a fan-forced draught taking the air from the ceiling or roof of the boiler- or engine-room, where it is often at a temperature above 100° F., makes a saving of from 10 to 20 per cent. of the coal over the waste from a low natural draught, besides the comfort of a modified temperature of the room. A fan should be large enough to furnish the required amount of air at as moderate a speed as will give the proper pressure, for the power to drive a fan increases as the cube of the speed.

A fan, the tips of its blades running at 65 feet a second, will give a draught of 1 inch of water. If possible, coal should not be burned with a stronger draught than this, for with a stronger draught the fires need very careful watching to prevent holes from burning through and letting in too much air, although mechanical stokers which have a constant-feeding attachment may have as much as 3 inches of draught without affecting the economy, for a strong draught will force the air through a heavy fire in more intimate contact with the fuel, and in this way be an aid to perfect combustion. A draught of $\frac{1}{2}$ inch of water is about as low as it is possible to obtain complete and smokeless combustion. With this amount of draught from 12 to 20 pounds of coal per hour per square foot of grate-surface may be burned.

CHAPTER VI

HEAT-ECONOMY OF THE FEED-WATER

THE saving of heat that would otherwise be wasted or lost by the exhaust-steam and the chimney-gases is of great consideration in the economy of steam-power. The exhaust-steam water-heater and the chimney-heat economizer are the only real saving devices that affect the cost of fuel. The live, steam heaters free the water from its incrusting elements, and injectors are only convenient mechanical substitutes. The following table shows the saving in percentage of the total fuel used by heating the feed-water between various initial and final temperatures:

TABLE XIII.—PERCENTAGE OF SAVING IN FUEL BY HEATING FEED-WATER. STEAM AT 70 POUNDS GAUGE-PRESSURE.

Initial-temperature feed.	TEMPERATURE TO WHICH FEED IS HEATED.														
	100°	110°	120°	130°	140°	150°	160°	170°	180°	190°	200°	210°	220°	250°	300°
35°	5.53	6.38	7.24	8.09	8.95	9.89	10.66	11.52	12.38	13.24	14.09	14.95	15.81	19.40	29.34
40°	5.12	5.97	6.84	7.69	8.56	9.42	10.28	11.14	12.00	12.87	13.73	14.59	15.45	18.89	28.78
45°	4.71	5.57	6.44	7.30	8.16	9.03	9.90	10.76	11.62	12.49	13.36	14.22	15.09	18.37	28.22
50°	4.30	5.16	6.03	6.89	7.76	8.64	9.51	10.38	11.24	12.11	12.98	13.85	14.72	17.87	27.67
55°	3.89	4.75	5.63	6.49	7.37	8.24	9.11	9.99	10.85	11.73	12.60	13.48	14.35	17.38	27.12
60°	3.47	4.34	5.21	6.08	6.96	7.84	8.72	9.60	10.47	11.34	12.22	13.10	13.98	16.86	26.56
65°	3.05	3.92	4.80	5.67	6.56	7.44	8.32	9.20	10.08	10.96	11.84	12.72	13.60	16.35	26.02
70°	2.62	3.50	4.38	5.26	6.15	7.03	7.92	8.80	9.68	10.57	11.45	12.34	13.22	15.84	25.47
75°	2.19	3.07	3.96	4.84	5.73	6.62	7.51	8.40	9.28	10.17	11.06	11.95	12.84	15.33	24.92
80°	1.76	2.65	3.54	4.42	5.32	6.21	7.11	8.00	8.8	9.78	10.67	11.57	12.46	14.81	24.37
85°	1.30	2.22	3.11	4.00	4.90	5.80	6.70	7.59	8.48	9.38	10.28	11.18	12.07	14.32	23.82
90°	0.89	1.78	2.68	3.58	4.48	5.38	6.28	7.18	8.07	8.98	9.88	10.78	11.68	13.82	23.27
95°	0.45	1.34	2.25	3.15	4.05	4.96	5.86	6.77	7.66	8.57	9.47	10.38	11.29	13.31	22.73
100°	0.00	0.90	1.81	2.71	3.62	4.53	5.44	6.35	7.25	8.16	9.07	9.98	10.88	12.80	22.18

The feed-water furnished to steam-boilers has to be heated from the normal temperature to that of the steam before evaporation can commence, and this generally at the expense of the fuel which should be utilized in making steam. This temperature at 75 pounds pressure is 320° F., and if we take 60° F. as the average temperature of feed, we have 260 units of heat per pound, which, as it takes 1.151 units to evaporate a pound from 60° F., represents 22.5 per cent. of the fuel.

All of this heat therefore which can be imparted to the feed-water is just so much saved, not only in cost of fuel but in capacity of boiler. But it is essential that it be done by heat which would otherwise be wasted. All heat imparted to feed-water by injectors and live-steam heaters comes from the fuel and represents no saving.

The number of square feet of surface required in feed-water heaters, for each horse-power, assuming an abundance of exhaust-steam is available, may be found by the following formula: $S = .227 \log_{10} \frac{T_s - T_1}{T_s - T_2}$, in which S = square feet of tube-surface per horse-power, or the surface required to heat 34.5 pounds per hour; T_s = temperature of the steam; T_1 = temperature of the water entering the heater, and T_2 = the temperature of the water leaving the heater. The horse-power of heater per square foot of surface is $1 \div S$. The result obtained by the use of the formula should be multiplied by 1.12 for brass tubes and by 1.67 for iron tubes.

Table XIV gives the tube-surface in square feet required to heat 34.5 pounds of water per hour, or for each boiler horse-power.

TABLE XIV.—AREA OF HEATING-SURFACE REQUIRED IN FEED-WATER HEATERS PER BOILER HORSE-POWER, 34½ POUNDS PER HOUR.

Initial temperature.	Temperature of Boiler-Feed. 170°			180°			190°			200°		
	Copper.	Brass.	Iron.	Copper.	Brass.	Iron.	Copper.	Brass.	Iron.	Copper.	Brass.	Iron.
50°	.15	.17	.24	.20	.23	.34	.22	.24	.36	.29	.32	.46
60°	.14	.16	.23	.19	.22	.33	.21	.23	.35	.28	.31	.45
70°	.13	.15	.22	.18	.21	.31	.20	.22	.34	.27	.30	.44
80°	.12	.14	.21	.17	.20	.29	.19	.21	.32	.26	.29	.43
90°	.11	.13	.19	.16	.18	.28	.18	.20	.30	.25	.28	.41
100°	.10	.12	.17	.15	.17	.26	.17	.19	.29	.24	.27	.40
110°	.09	.10	.16	.14	.16	.24	.16	.18	.27	.23	.26	.38
120°	.08	.09	.14	.13	.15	.22	.15	.17	.25	.22	.24	.36
130°	.07	.08	.12	.12	.13	.20	.14	.15	.23	.21	.23	.34

The extent of heating-surface given in the table for 1 horse-power will generally be found ample for coil-heaters and for horizontal double-flow water-tube heaters having copper coils and copper tubes respectively, and for vertical water-tube heaters with corrugated copper tubes when the aggregate tube-area corresponding to the direction of flow is not large, thus insuring a more rapid circulation.

FEED-WATER HEATERS

Feed-water heaters are made of both the closed and open types. In the closed type the water is caused to circulate through tubes arranged in different ways, while the exhaust-steam envelops the tubes from end to end in passing from the inlet to the outlet. In other heaters of the closed type the water is outside and the steam inside the tubes. The former method of heating, however, is the better, because a positive and more rapid circulation of the water is thus secured; and it has been found that the efficiency of a feed-water heater depends largely upon proper circulation, the absorption of heat taking place more rapidly with a brisk circulation than when the circulation is sluggish. Horizontal tubes frequently give somewhat better results for a given area of heating-surface than do vertical tubes, but the slight loss due to the position is fully compensated for in the vertical types by allowing slightly more surface and causing the water to flow from top to bottom against the current of steam, so that it makes but little difference in practice whether a horizontal or vertical heater is selected, as far as being able to heat water to the desired temperature is concerned.

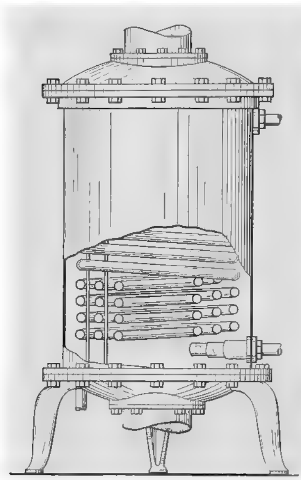


FIG. 73.—Multicoil-heater.

When certain constructions of closed heater are employed, notably the coil, a somewhat smaller heater may be used, owing to the positive and rapid circulation and the efficient form and arrangement of the heating-surfaces. The closed heater is also adapted to use with condensing-engines. Either type may be successfully employed in connection with heating systems, the shells being made of ample strength to resist the pressures generally employed in heating by exhaust-steam. The temperature of the water leaving the heater is usually about the same with both the closed and open types under the same conditions. The shells and pipes should be covered with some non-conducting material to prevent radiation, so that as much

steam as possible may be provided for other purposes when the steam is to be utilized after passing through the heater.

The open heater is not necessarily subjected to any pressure either of steam or water except that due to the weight of the water it contains. This type of heater furnishes a settling-chamber for the impurities in the feed-water, which with muddy water, or water containing large quantities of other impurities, is of great advantage. By introducing suitable trays or pans a considerable quantity of scale-making material may also be removed, while the condensation of a portion of the steam furnishes a certain amount of pure water, which is added to that in the heater. One of the greatest difficulties formerly experienced with open heaters was found in avoiding the effects of the cylinder-oil carried into the heater by the exhaust-steam. In but few cases at present is this difficulty experienced, the construction being such as to exclude the greater part of the oil, and to give the water sufficient time in the heater to permit the remaining oil to rise to the surface of the water and be drained off through suitable waste-pipes. When selecting an open heater it is important to investigate the provisions made for disposing of the oil and preventing it from entering the boiler. The open heater should be so constructed as to permit easy and frequent cleansing, when necessary, and the ready removal of the filtering material and of pans or trays when these are employed.

The open feed-water heater is designed so that the water entering at the top will be finely divided and will fall through the reservoir of steam in the form of a fine spray or a very thin film, thus bringing practically all the water into close contact and causing an intimate commingling with the steam in the shortest time possible, that is when considering the total time required for the water to pass through the heater. The time during which the water is passing downward from the inlet to the water-reservoir at the bottom is, however, made as long as possible so as to secure the thorough absorption of the heat in the steam, for upon the thoroughness of this process depends the temperature to which the water can be heated with a given initial temperature and a given temperature of steam.

In Fig. 74 is a sectional view of the Berryman heater, in which the exhaust-steam passes through the inverted U-shaped tubes, which

permits both ends of the tubes to be expanded into the same tube-sheet, and thus they cannot be affected by expansion or contraction. Each tube is absolutely independent of every other tube. They seldom coat or scale, and thus their full heating-surface is indefinitely maintained.

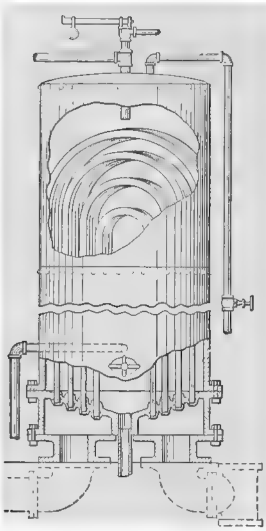


FIG. 74.—Berryman heater.

The head into which the tubes are set is cast-iron, from 2 to 3 inches in thickness. The holes for receiving the tubes are first drilled the size of the inside of the tubes, then counterbored to within ½ inch of the bottom of the tube-head, leaving a solid shoulder on which the tube rests. A groove is cut in the centre of the bore of the thickness of the tube. The U-shaped tubes are then expanded in these grooves, their shape preventing any strain from expansion or contraction, which, together with the manner of setting them, prevents leaking or getting loose.

The tube-head is concave, and at its lowest point a mud blow-off is arranged, through which the sediment and other impurities can be removed. It should be opened for a few seconds as often as the condition of the water necessitates, which can readily be determined by experiment.

The exhaust-steam enters at one side of the heater, passes up through the tubes and down and out on the other side. The ports in the heater may be arranged to meet the needs in any case.

The water enters the heater at the side, but at a sufficient distance from the bottom to prevent disturbing the sediment which has collected.

The water leaves the heater through a pipe which extends down a few inches from the top, and is thus taken at the hottest part.

The Wainwright heater, Fig. 75, has the advantage derived from the tubes being corrugated, which not only gives increased surface to the tubes, but makes them elastic, and thus insures their tightness in the tube-heads. There is an ample settling-chamber at the bottom and a surface blow-off and storage-room at the top. The tubes

occupy only one-fourth of the shell-area, and the net shell-area is at least three times as large as the area of the exhaust-pipe. This allows the steam to flow freely about the small columns of moving water.

The water-chambers are divided into several compartments, and the partitions are so arranged that they direct the flow of the feed-water back and forth through the heater, using the various groups of tubes in succession, with a consequent increase in velocity over that obtained in the non-return type of heater. Each of these groups of tubes contains a sufficient number of tubes to give a sectional

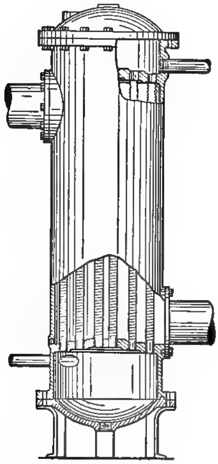


FIG. 75.—Wainwright heater.

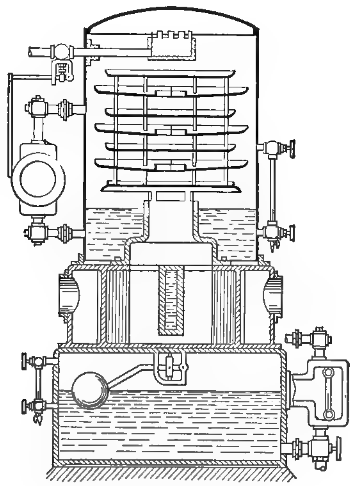


FIG. 76.—Cookson heater, purifier, and oil-separator.

area which is at least twice the sectional area of the feed-pipe. This increase in the speed of the feed-water brings all parts of it into contact with the heating-surface and insures a uniform use of all the tubes. Experiments have shown that in some constructions of multi-tubular heater the water may remain almost stagnant in a portion of the tubes. The practical result is that there is offered in the even flow a heater with which there can be a very high final temperature, approximating 212° F. under ordinary conditions of exhaust with non-condensing-engines.

In the Cookson heater, Fig. 76, the steam enters the side and strikes the V-shaped, oil-separating plates which divide the volume of steam,

the ribs on the plates catching the oil and moisture in the steam. The steam then enters the enlarged portion of the exhaust-tube, where it passes into the opposite expansion and oil-separating chamber and discharges into the atmosphere or heating system. At the top of the heater is a vent-pipe for carrying off the air relieved from the water in heating. The vent-pipe is to be connected with the exhaust-outlet. The cold-water supply enters in a spray and condenses the steam, forming a partial vacuum, which draws the required amount of steam to heat the water through the large tube in the centre. Only that amount of steam necessary to heat the water comes in contact with it, the remainder passing on to the heating system or the atmosphere.

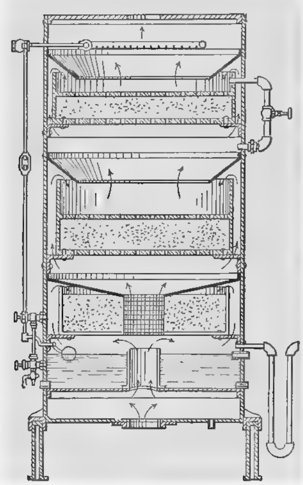


FIG. 77.—Feed-water heater and filter.

The water-supply is connected with the water-inlet valve, which is opened and closed by the water-regulator, maintaining at all times a uniform water-level in the heater. The water entering the spray-box at the top of the heater overflows in a spray to the pan below, and, overflowing this pan, falls in a spray into the next. The water passes from this third pan over its outer edge, following down on the under side to the next pan below, and so on down. The last pan is bolted to the top of the exhaust-tube. The water sprays from this last pan to the water below. All pans, with the exception of the bottom one, are loose, made in halves, and are readily removed through the man-hole. The object of these pans is to catch the lime deposits. The water, after having been heated in direct contact with the steam, enters the hollow partition at the back of the exhaust-tube. The water discharges from the hollow partition near the front into the filtering-chamber below, where the remaining impurities in suspension are removed by filtration. The filtering-chamber is filled with coke or excelsior, and at the back of this chamber is a perforated plate preventing the filtering material from passing through to the pump. A strainer-plate is also placed at the blow-off connections. The blow-off and oil-discharge pipes are placed on the side opposite

the exhaust-inlet. The two oil-separating chambers are connected by a small opening through the hollow partition at the bottom, through which the oil and condensed steam drain, passing from there into the oil-discharge pipe and thence to the sewer.

In the feed-water heater and filter, Fig. 77, the exhaust-steam enters at the bottom and flows into the first compartment through a short pipe, thence through the annular opening surrounding the second compartment into the latter, thence through another annular opening into the next compartment. After passing through the annular openings the steam comes in contact with baffle-plates, which direct the steam through the falling water and condense it. A ring-pipe at the top distributes the water upon a baffle-plate, from which it falls upon the top filter and so on through the three filter sections.

The Hoppes standard feed-water heater, shown in Fig. 78, is supplied with pans of the same design as those in the live-steam, feed-water purifier. The water in flowing over the sides and bottoms

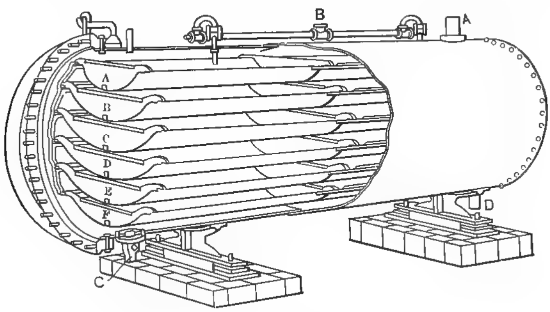


Fig. 78.—Hoppes feed-water heater.

of the pans comes in direct contact with the steam and is heated nearly to the temperature of the exhaust-steam.

This heater is especially designed to be used where the water is bad, and one peculiar advantage is had in the fact that the water flows along the under side of the pans, or the lime formation thereon, and thus comes in direct contact with the exhaust-steam, no matter how thick the lime formation may be on the pans. The apparatus is provided with a large oil-catcher, located in the rear, and through which all the steam passes and is purified before entering the heater.

A float is provided which operates a balanced valve for the regulation of the feed-water. The entire front head is easily removed and swung to one side by a crane provided for this purpose, so that the pans may be readily removed. As the pans contain all of the lime and other solids formed in the heater, the entire work of cleaning is performed outside of the heater.

THE GREEN FUEL-ECONOMIZER

This apparatus consists of a stack of tubes arranged vertically in the flue leading from the boiler to the chimney (as illustrated in Fig. 79), and is designed to utilize the waste heat in the gases passing off from the furnace. This is accomplished by absorbing the low-tem-

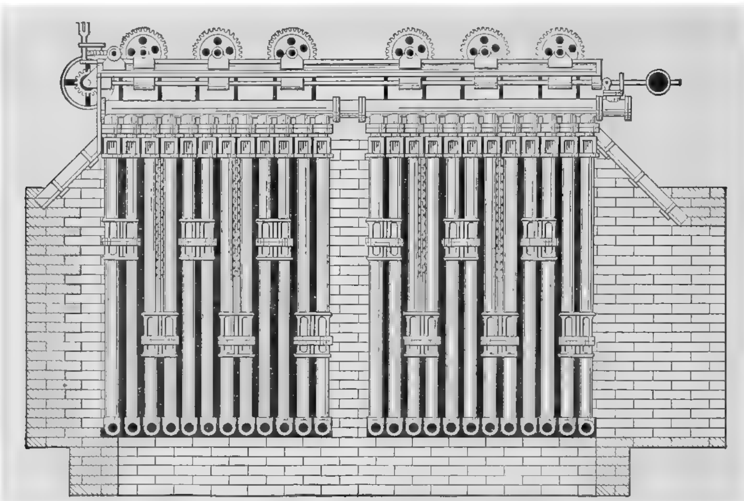


FIG. 79.—Green fuel-economizer.

perature heat of the gases in heating the feed-water, which is pumped through the economizer before entering the boiler. The waste gases are led to the economizer by the ordinary flue from the boilers to the chimney.

The feed-water is forced into the economizer by the boiler feed-pump, or an injector, at the lower branch pipe nearest the point of exit of the gases, and emerges from the economizer at the upper branch pipe nearest the point where the gases enter.

Each tube is provided with a geared scraper, which travels continuously up and down the tubes at a low rate of speed, the object being to keep the external surface clean and free from soot.

The mechanism for working the scrapers is placed on the top of the economizer, outside the chamber, and is very simple and effective; the motive power is supplied either by a belt from some convenient shaft or by a small independent engine or motor. The power required for operating the gearing is very small.

The apparatus is fitted with blowoff- and safety-valves, and a space is provided at the bottom of the chamber for the collection of the soot removed by the scrapers.

The scrapers are three in number and encircle the pipes with the joints overlapping one another. They have thin, beveled cutting-edges which entirely remove any accumulation of soot. Under conditions where a forced circulation may be an advantage, circulating blow-off manifolds are introduced. By means of these manifolds any portion or the whole of the economizer can be made to circulate, and at the same time every section can be thoroughly blown off. As the economizer should be blown off for a few moments at least once a day, the valves are connected together by a long lever, which makes the operation very simple and takes the least possible time to operate.

CHAPTER VII

THE INJECTOR AND THE STEAM-PUMP

THE injector and its theory were matters of much discussion during the early years of its use, and its final solution has been mathematically demonstrated as the elimination of the volume of the steam-jet at a high velocity by the instantaneous absorption of its latent heat in contact with the incoming water, thus imparting its velocity momentum to the water around it, by which interchange of temperatures the volume of the steam is reduced to the volume of its water-base. By this action its proportionate velocity is imparted to the incoming annular water-jet, which becomes a solid water-jet at the end of the combining-nozle, the momentum of which is far greater than is required to overcome the resistance of the boiler-pressure, and the jet crosses a starting relief-space and enters the delivery-nozle, opening by its force the boiler check-valve.

The formulas representing the action of an injector are as follows: For the velocity of the injection at the exit of the combining-nozle we have, $V = 12.19\sqrt{p}$ in feet per second, in which p = the gauge-pressure. The volume of water and condensed steam passing the nozle of the combining-tube, per second, will be $0.016\ (W + W_0)$; and if A be the area and W the weight of the steam, and W_0 the weight of the water, then $\frac{A = 0.016\ (W + W_0)}{V}$; in which V = velocity, as found above.

The velocity of the steam may be found from the formula: $V = 23.2687\sqrt{p\ v\left(1 - \left(\frac{p_2}{p}\right)0.1189\right)}$, in which p = absolute initial pressure, v = volume of steam at initial pressure, and p_2 = pressure in the chamber between the nozles—generally atmospheric pressure.

Table XV is an approximate service of a simple injector, equal to the delivery of about 1 pound of water per second at a temperature of 160° F. from feed-supply at 60° F.

TABLE XV.—GAUGE-PRESSURES, NOZLE-DIAMETERS, AND VELOCITIES OF STEAM AND WATER AND THEIR RATIOS.

Gauge-pressure, pounds.	Diameter steam-nozle, inches.	Diameter water-nozle, inches.	Velocity steam, feet per second.	Velocity steam and water, feet per second.	Ratio of velocity steam to water.	Ratio of weight, water to steam.	Ratio of volume, steam to water.
30	0.28	0.21	2007.9	66.7	30.	10.3	55.9
40	0.24	0.20	2178.8	77.1	28.	10.3	46.2
50	0.22	0.19	2213.5	86.2	25.	10.4	39.4
60	0.20	0.18	2428.8	94.4	25.	10.5	34.4
70	0.18	0.178	2522.3	101.2	25.	10.5	30.4
80	0.17	0.172	2554.1	108.0	24.	10.5	27.6
90	0.167	0.166	2590.6	115.6	22.	10.5	25.2
100	0.159	0.160	2735.8	121.8	22.	10.5	22.8
120	0.142	0.154	2842.7	133.5	21.	10.6	19.6
140	0.133	0.149	2922.3	144.2	20.	10.6	17.2
160	0.127	0.143	2999.7	154.2	19.	10.6	15.3

Under ordinary conditions an injector will feed about 12 pounds of water to a boiler per pound of steam, or 13 pounds including its own weight.

The limit of the feed-water temperature for an injector is about 110° F., so that open feed-water heaters cannot supply the water; but injectors can feed boilers through closed heaters to advantage, with possibilities of raising the temperature of the feed-water to near 212° F.

Of the many models of injectors on the market, the tandem and double combining-tube models are taking the lead for efficiency and reliability. Following are illustrated some of the various models in section, showing their details of construction:

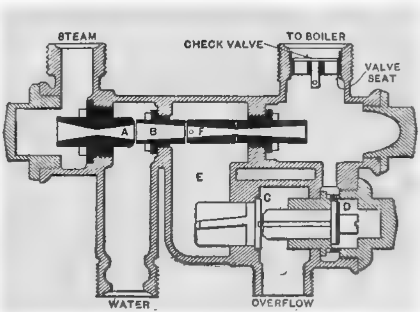


FIG. 80.—Penberthy injector.

The Penberthy injector, Fig. 80, special model, has three fixed nozle-tubes in line. The opening of a detached valve gives steam to the chamber E through the annular orifice between the combining- and receiving-nozles at F, and by its pressure opens the relief check-valves C and D. When the water-current is established, the pressure in the chamber next to the boiler check-valve closes the check-valve

D, which by the contact of its wings with the check-valve C closes it, and the full pressure opens the boiler check-valve.

The Little Giant, Fig. 81, is an adjustable injector in which two of its three tubes are fixed. The combining-tube is movable for adjustment by the lever-handle, which by drawing the combining-tube toward the steam-nozle regulates the flow of water, and the steam is regulated by a detached valve. The relief check-valve C automatically closes on the establishment of the water-current.

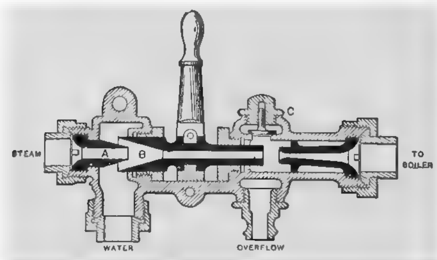


FIG. 81.—Little Giant injector.

The Lunkenheimer injector, Fig. 82, has four fixed nozle-tubes, with all the valves required for operating it attached to the injector. The steam-regulating valve is adjusted by a lever as shown; D is the stop-check to the overflow, which is carried around the body of the injector to the nozle below. In starting, the pressure, by the escape of steam at the annular orifice into the chamber E, opens the relief-check C. When the water-current is established, the overflow-check at D is closed, and the pressure from the nozle of the second section of the combining-tube in the chamber S closes the check-valve C, and the water and steam pass this gap in a solid stream.

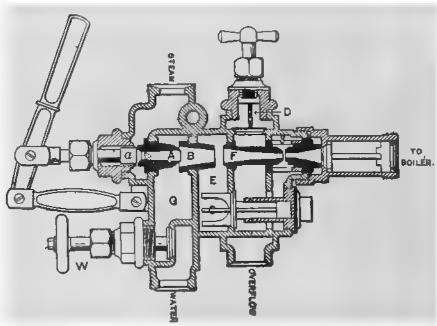


FIG. 82.—Lunkenheimer injector.

Of the tandem nozle-injectors there are a great variety of models on the market, each having its own peculiar features. The double-tube injectors, although seemingly somewhat more complex in their construction, are claimed to deliver the feed-water at a higher temperature by the fact that the water passes successively through two combining-nozles.

As an example of this class we illustrate in section, in Fig. 83, the Metropolitan double-tube injector.

The steam is turned on from a separate valve; the first movement of the handle opens the first section of a double-beat valve at b, and gives the steam to the lifting-nozle A; the overflow passing freely through the check-valve C and the open valve at D. A further movement of the handle opens the second section of the double-beat valve B, and closes the overflow-valve D, when the flow of warm water from the first tube, M, flows into the chamber F, and to the second tube, and through the chamber G to the boiler. The pressure in G at the moment of discharge of the second tube closes the overflow-valve C.

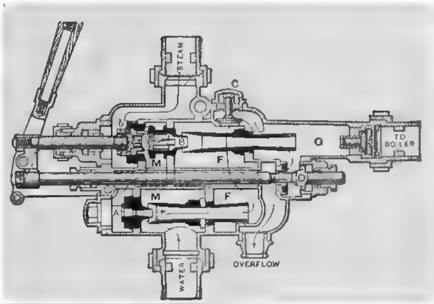


FIG. 83.—Metropolitan injector.

The Korting injector, Fig. 84, is of the double-tube variety, with an automatic movement by which the difference in area of the valve-disks at A and B allows the balance-lever to open the lifting-nozle first, and by a further movement of the handle opens the force-nozle B. The overflow is self-adjusting for both nozles.

The real efficiency in the injector, and its economy in saving part of the heat lost by the exhaust, are found in the exhaust-injector

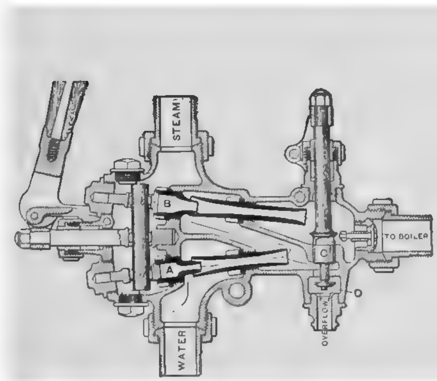


FIG. 84.—Korting injector.

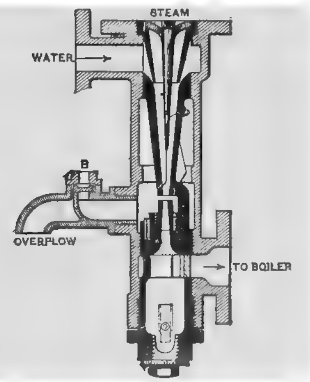


FIG. 85.—Exhaust-injector.

(shown in Fig. 85) of the triple-tube model, in which the centre or combining-tube has a hinged section which opens automatically by

the incoming exhaust, and allows a free flow to draw the water into the nozle and through the overflow. When the water-current is established the hinged section of the combining-tube automatically closes, and the injector operates the same as others for feeding a boiler. The portion of the exhaust not used by the injector may pass through a heater which the injector feeds, thus increasing the feed-water temperature.

The efficiency of the injector as a heat device is claimed to be theoretically perfect, as it returns all the heat it receives from the boiler save the radiation and the small losses in starting; but as a pump for elevating water its efficiency is very low in comparison with the steam-pump, being about one-fifth as efficient. The work of forcing water into a boiler, say at 80 pounds pressure, in the proportion of 13 pounds of water to 1 pound of steam, as shown in Table XV, is, $144 \times 80 \times 13 \times 0.016 = 2{,}396$ foot-pounds. One pound of steam in the direct-acting steam-pump will, at 80 pounds boiler-pressure, do the actual work of 10,000 foot-pounds, or over four times as much as an injector. A pump feeding a boiler at 80 pounds pressure which generates $8\frac{1}{2}$ pounds of steam per pound of coal consumes about 2 per cent. of the fuel.

THE STEAM-PUMP AND ITS WORK

The power required to force water against a given pressure or height must include in its resistance the height of the draught or suction and the friction of the pump as a machine, as the three static elements against which the pump must work; and also the element of action to keep the pump moving at the required speed. The friction and action elements of pump-work, especially in small pumps, may be as much as 60 per cent. greater than the total static force of the pump's work.

In pumps used for boiler-feeding with pressure-supply, the usual ratio of diameter of steam-cylinder to water-cylinder is from 1.20 to 1.25; but where extreme suction-lift has to be overcome, a ratio of 1.30 is a safer assurance of proper action, and in such cases only pumps with very small clearance can be relied upon.

The formulas for the balance of pressure and areas in steam-pumps, to which should be added the necessary steam-pressure for actuating the pumps, are:

$$\frac{\text{water-pressure}}{\text{area steam-cylinder} \div \text{area water-cylinder}} = \text{steam-pressure.}$$

$$\frac{\text{area water-cylinder} \times \text{water-pressure}}{\text{area steam-cylinder}} = \text{steam-pressure.}$$

$$\frac{\text{water-pressure}}{\text{steam-pressure}} = \frac{\text{area steam-cylinder}}{\text{area water-cylinder}}$$

$$\frac{\text{area steam-cylinder} \times \text{steam-pressure}}{\text{water-pressure}} = \text{area water-cylinder.}$$

$$\frac{\text{area steam-cylinder} \times \text{steam-pressure}}{\text{area water-cylinder}} = \text{water-pressure.}$$

For obtaining the actual horse-power that is required to operate a pump, we have weight of water in pounds per minute $\times$ height (or pressure $\times 2.3) \div 33{,}000 =$ horse-power.

The decreasing pressure of the atmosphere at a height above sea-level materially affects the suction-lift of a pump. Assuming that the practical lift of a pump at sea-level is 25 feet, the following table shows the comparative height, pressures in pounds, and equivalent head of water in feet, and the corresponding practical lift of pumps:

Table XVI.—Height and Atmospheric Pressure, with Equivalent Head of Water and Pump-Lift.

Altitude Above Sea-Level.	Pressure, pounds per square inch.	Equivalent head of water, feet.	Practical suction-lift in feet.
At sea-level	14.70	33.95	25.
¼ mile = 1,320 feet	14.02	32.38	24.
½ " = 2,640 "	13.33	30.79	23.
¾ " = 3,960 "	12.66	29.24	21.
1 " = 5,280 "	12.02	27.76	20.
1¼ " = 6,600 "	11.42	26.38	19.
1½ " = 7,920 "	10.88	25.13	18.
2 " = 10,560 "	9.88	22.82	17.

In the ordinary practice of piping pumps for feeding boilers the friction of the water in the pipes is not considered; but sometimes long suction-pipes are required, when the friction may be serious, or an obstacle to high lifts. Five hundred to 1,000 feet are feasible distances for pump-suction with an ample air-chamber on the suction-pipe near the pump and with lifts as in the table, less the friction-head for pipe and fittings.

The formula for straight pipe is: $\frac{L}{d} \times \frac{4V^2+5V-2}{1,200}$ = friction-head in feet.

L = length in feet; d = diameter in inches; V = velocity of the water in feet per second. An elbow is equal to 60 diameters, and a globe-valve equal to 90 diameters, of the pipe, and should be added to the length of the pipe.

Of the many models of boiler-pumps, we can illustrate only a few of those having special features.

In Fig. 86 is shown a sectional view of the Knowles steam-pump. Freedom from stoppage on a dead centre of the valve-movement is

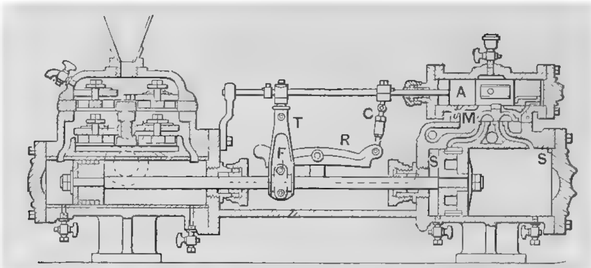


FIG. 86.—Knowles single pump.

secured by the use of the auxiliary piston A, which works in the steam-chest and drives the main slide-valve M. This main valve is of the B form and moves on a flat seat; it has on top a stem which fits into a recess in the piston A. The chest-piston A has a slight rotation from the curved rocker-bar R, which alternately covers and uncovers

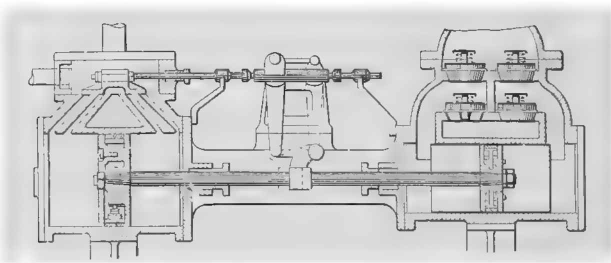


FIG. 87.—Knowles duplex pump.

small ports, S, S, which enter the cylinder at each end near the head. The steam-piston runs over the main ports, and by its cushion operates the piston-valve and the main valve.

The Knowles duplex pump is shown in section in Fig. 87. This pump has a double set of steam-ports which produce a cushion at each piston-stroke by covering the inside ports alternately; the plain D valve making the closure by its movement. A rocker-arm linked to the piston-rod of each side of the pump operates the opposite valve.

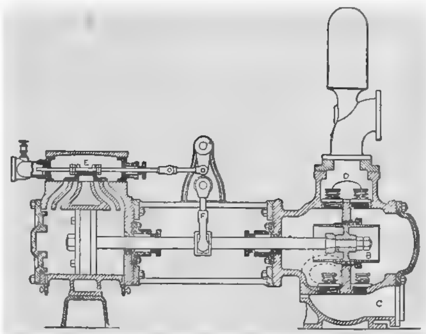


FIG. 88.—Worthington duplex pump.

The Worthington duplex pump, Fig. 88, has the same valve-movement and cushioning-ports as above described; but the water-piston is of the plunger form, with the inlet-valves at the bottom of the cylinder.

Fig. 89 shows the vacuum-pump and jet-condenser, and Figs. 90 and 91 show the details of the valve-gear used on the Deane single-cylinder steam-pumps. The main valve is operated by a small piston called the valve-piston. The ears on the main valve fit tightly in a slot cut in the valve-piston, so that when the valve-piston moves in either direction it carries the main valve with it.

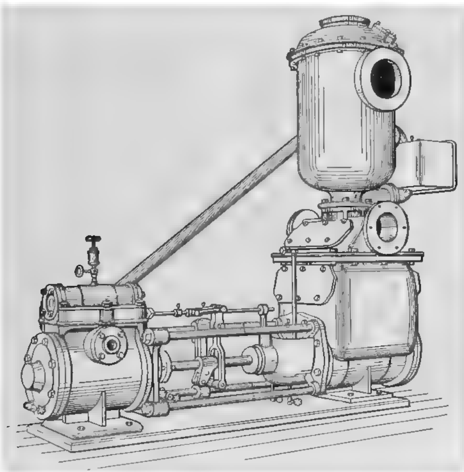


FIG. 89.—Deane vacuum-pump with jet-condenser.

The valve-piston is fitted to and slides in a cylindrical bore in the valve-chest, and is actuated by steam admitted to the opposite ends of the chest. The admission and exhaust of this steam are controlled by a secondary valve, which admits or exhausts the steam for the valve-piston through the small ports at the sides of the cylinder and chest. The secondary valve derives its motion, through the valve-rod, tappets, and links shown, from the

main piston-rod. Thus, the movement of the secondary valve, and hence the valve-piston and main valve, are controlled by the main

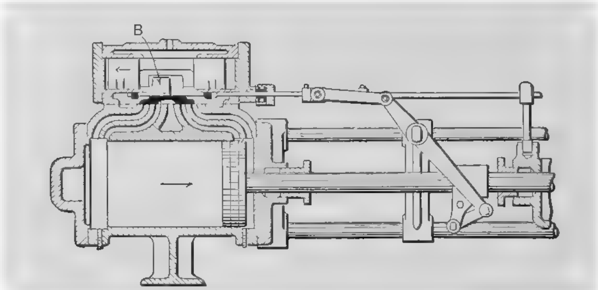


FIG. 90.—Valve-gear, Deane pump.

piston. The valve-piston, it will be noticed, has a steam-jacket which insures equal expansion of the parts and prevents binding.

The piston-rod arm is fastened to the piston-rod, and through the connection of lever and links its motion causes the tappet-block to slide back and forth on the valve-rod between the two tappets. These tappets are keyed to the valve-rod so that when the tappet-block strikes either tappet it carries with it the valve-rod and secondary valve. When the piston moving in the direction indicated by the arrow has come almost to the end of the stroke, the tappet-block comes in contact with the left-hand tappet, and the further movement of the piston throws the secondary valve to the left until the edge A, Fig. 91,

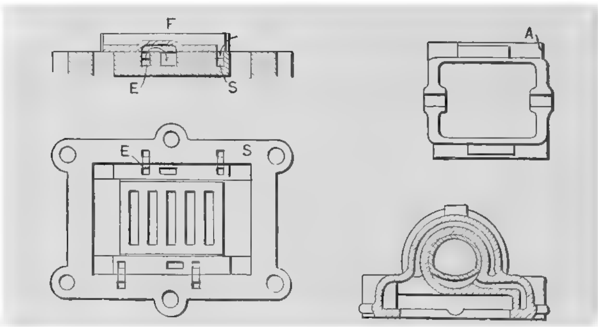


FIG. 91.—Valve-chest and auxiliary valve.

uncovers the small port S. The port S, together with passages in the cylinder- and valve-chest, allows the steam to fill the space between the right-hand end of the valve-piston and the valve-chest head,

and exerts a pressure forcing the valve-piston in the direction indicated by the arrow. In the illustration, Fig. 90, the valve-piston has already moved part of the way, carrying the main valve with it far enough to partially open the steam-port which admits steam to the right-hand end of the cylinder, and the main piston is ready to start back in the other direction. The port E and the chamber F in the secondary valve, as shown in Fig. 91, provide for the exhaust of steam from behind the left-hand end of the valve-piston in the same manner and at the same time that steam is admitted behind the right-hand end. The location of the exhaust-ports in the chest is such as to allow for proper cushioning of the valve-piston to prevent its striking the heads. The small ports on the other side of the steam-cylinder control the motion of the valve in the other direction, and act in exactly the same manner. In case the steam-pressure should for any reason fail to start the valve-piston at the proper time there is a lug, B, Fig. 90, provided on the valve-rod which comes in contact with the valve-piston and brings to bear the whole power of the steam-cylinder to start it. It is readily seen that the correct timing of the valve-movements is independent of the position of the tappets. If they are too near together the valve will be thrown too soon, and thus the stroke of the pump will be shortened; while, on the other hand, if they are too far apart, the pump will complete its stroke without moving the valves.

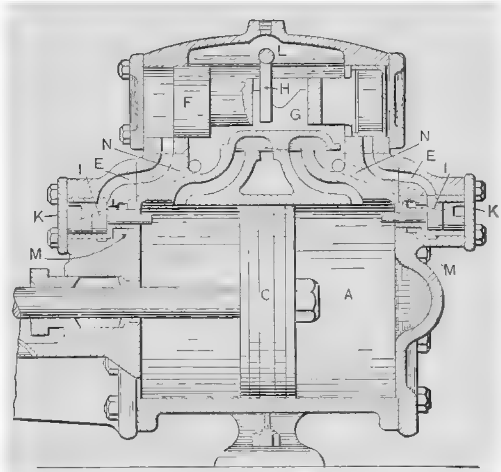


FIG. 92.—Sectional view of Cameron pump.

In the Cameron pump the plunger is reversed by means of two plain tappet-valves, shown in Fig. 92, and the entire mechanism thus consists of four pieces only, all working in direct line with the main piston. It is simple and without delicate parts.

A is the steam-cylinder; C, the piston; L, the steam-chest; F, the

chest-plunger, the right-hand end of which is shown in section; G, the slide-valve; H, a lever, by means of which the steam-chest plunger F may be reversed by hand when expedient; I, I are reversing-valves; K, K are the reversing-valve chamber-bonnets, and E, E are exhaust-ports leading from the ends of the steam-chest direct to the main exhaust by means of passages, M, M, which lead directly thereto, although the connection is not shown, being cut away in the sectional view, and closed by the reversing-valves I, I.

The piston C is driven by steam admitted under the slide-valve G, which as it is shifted backward and forward alternately connects opposite ends of the cylinder A with the live-steam pipe and exhaust. This slide-valve G is shifted by the auxiliary plunger F; F is hollow at the ends, which are filled with steam, and this, issuing through a hole in each end, fills the spaces between it and the heads of the steam-chest in which it works. Pressure being equal at each end, this plunger F, under ordinary conditions, is balanced and motionless; but when the main piston C has travelled far enough to the left to strike and open the reversing-valve I, the steam exhausts through the port E from behind that end of the plunger F, which immediately shifts accordingly and carries with it the slide-valve G, thus reversing the pump. No matter how fast the piston may be travelling, it must instantly reverse on touching the valve I. In its movement the plunger F acts as a slide-valve to close the port E, and is cushioned on the confined steam between the ports and steam-chest cover. The reversing-valves I, I are closed, as soon as the piston C leaves them, by a constant pressure of steam behind them, direct from the steam-chest through the ports N, N, shown by the dotted lines.

In the McGowan single-cylinder pump the main valve is of the B form and is driven by a chest-piston or valve-driver, as shown in Fig. 93. Steam is alternately admitted through one of the cavities in the valve and is released through the other, the central port in the valve-seat admitting the live steam. Immediately below the ends of the steam-chest are two tappet-valves, which normally cover the auxiliary ports (shown by dotted lines), leading to the ends of the steam-chest and connecting the latter with the main exhaust-ports. The tappet-valves are raised by means of levers, the ends of which project downward and into the cylinder, so that when the piston nears

the ends of the stroke it comes into contact with the levers and raises them slightly, the movement being merely sufficient to unseat the tappet-valves.

The tappet-valve levers are pivoted on a pin in a recess near the main ports, the latter being indicated by dotted lines.

When the piston reaches the end of the stroke, one of the tappet-levers is raised slightly and the corresponding valve is raised from its seat. This opens the port leading from the end of the steam-chest to the main exhaust-port and permits steam to escape into the latter. The pressure is thus lessened on one end of the chest-piston or valve-driver, and the steam pressing on the opposite end forces the valve-driver to the opposite end of its stroke, thus reversing the distribution of steam to the cylinder and starting the piston on the return stroke. The chest-piston is caused to move back and forth by live steam, the ends of the steam-chest being filled with steam at initial pressure. Permitting steam to escape from one end of the steam-chest causes a difference of pressure on the two ends of the chest-piston, which difference represents the propelling force that moves the main valve. The tappet-valves have a very slight lift, so that they operate without shock or noise and with the minimum of wear. The main valve is connected with the chest-piston or valve-driver in such a manner that all lost motion and wear is taken up automatically.

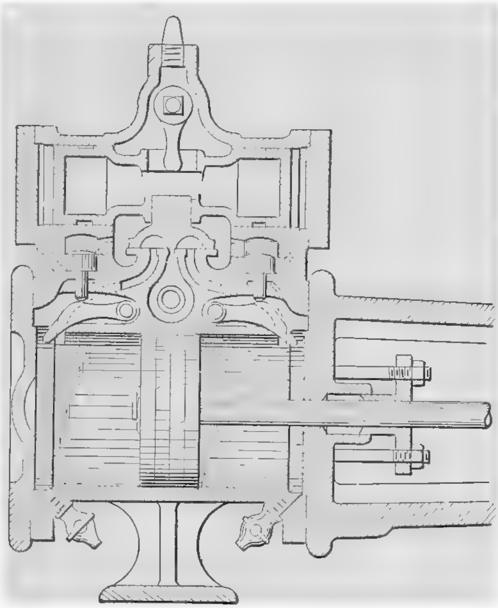


FIG. 93.—Sectional view of steam-end, McGowan pump.

A short rocker-shaft, extending through the steam-chest and at right angles to the valve-travel, carries a toe which depends in a slot in

the top of the chest-piston, so that in event the latter should chance to stop with the ports closed, the valve can be moved by hand without disconnecting any part of the pump.

THE GUILD & GARRISON PUMP

The steam-chest of this pump is a rectangular chamber, provided at each end with suitable cylinders to receive the pistons of the valve-driver E, Fig. 94. At the side of the valve-driver E, and in the steam-chest, is an auxiliary slide-valve, G, Figs. 94 and 95, which admits and releases the steam from the ends of the valve-driver. The valve-driver E has two slots at the centre, the lower one receiving the lug on the back of the main valve, and the upper one the toe on the rocker-shaft D. The rocker-shaft has two toes, the larger one engaging with the valve-driver, and the smaller one with the auxiliary slide-valve G, as shown in Fig. 96. Both the main and the auxiliary valves are plain slide-valves, so fitted as to take up wear automatically. The pendulum-lever J, Fig. 96, causes the shaft D to rotate, and by means of the toes previously referred to the valves are caused to move in unison.

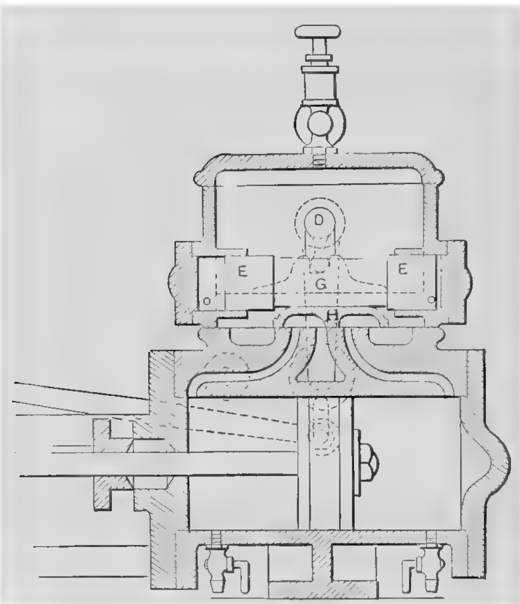


FIG. 94.—Steam-cylinder, Guild & Garrison pump.

The auxiliary valve G is in every respect similar to the slide-valve of an engine, and admits and releases the steam to and from the ends of the valve-driver.

The operation of the valves is as follows: The piston being at the end of the stroke, steam is admitted by the main valve, and the piston

moves toward the opposite end of the stroke. The two valves are also moved in the same direction by means of the rocker-shaft and the toes. This movement is continued until the piston has nearly completed the stroke, when the auxiliary valve opens one of the small ports leading to the end of the valve-driver, thus admitting steam at one end and releasing it from the other, which causes the valve-driver to move from one end of the steam-chest to the other, which movement also shifts the position of the main valve and reverses the motion of the main piston. The valve-driver is moved the greater part of the distance by means of the toe on the rocker-shaft, the stroke or travel of the driver being completed, thus reversing the steam-distribution by steam-pressure, which brings the opposite end of the slot in the driver in position to be again engaged by the toe on the rocker-shaft for the return stroke.

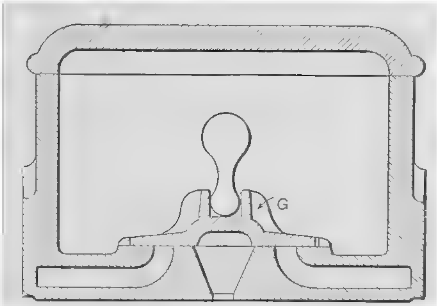


Fig. 95.—Sectional view of valve-chest, Guild & Garrison pump.

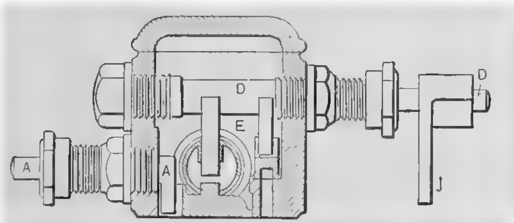


Fig. 96.—Auxiliary valve, Guild & Garrison pump.

In the Blake single-cylinder pump the main valve, which controls the admission of steam to and the escape of steam from the main cylinder, is divided into two parts, one of which, G, Figs. 98 and 99, slides upon a seat on the main cylinder, and at the same time affords a seat for the other part, D, which slides upon the upper face of G. As shown in the illustrations, D is at the left-hand end of its stroke, and G at the

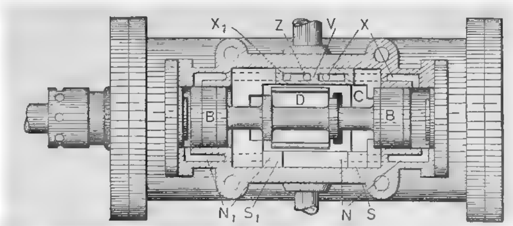


Fig. 97.—Plan of Blake pump-valve.

opposite or right-hand end of its stroke. Steam from the steam-chest J is therefore entering the right-hand end of the main cylinder through the ports E and H, and the exhaust is escaping through the ports H_1, E_1, K, and M, which causes the main piston A to move from right to left. When the piston has nearly reached the left-hand end of the cylinder the valve-motion moves the valve-rod P, and thus causes G, together with its supplemental valves R and S, S_1, Fig. 99 (which form, with G, one casting), to be moved from right to left. This causes steam to be admitted to the left-hand end of the supplemental cylinder, whereby the piston B will be forced toward the right, carrying D with it to the opposite or right-hand end of its stroke; for the movement of S closes N, the steam-port leading to the right-hand end, and the movement of S_1 opens N_1, the steam-port leading to the opposite or left-hand end, at the same time the movement of V opens the right-hand end of this cylinder to the exhaust, through the exhaust-ports X and Z. The parts G and D now have positions opposite to those shown in the cuts, and steam is therefore entering the main cylinder through the ports E_1 and H_1, and escaping through the ports H, E, K, and M, which will cause the main piston A to move in the opposite direction, or from left to right, and operations similar to those already described will follow when the piston approaches the right-hand end of the cylinder. By this simple arrangement the pump is rendered positive in action; that is, it will start and continue working the moment steam is admitted to the steam-chest.

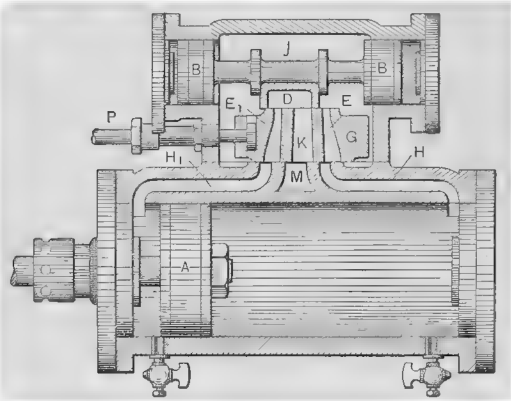


Fig. 98.—Section of Blake pump.

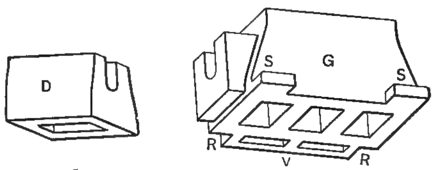


Fig. 99.—Main valve and rider.

The main piston A cannot strike the heads of the cylinder, for

the main valve has lead; or, in other words, steam is always admitted in front of the piston just before it reaches either end of the cylinder, even though the supplemental piston B be tardy in its action and remain with D at that end toward which the piston A is moving, for G would be moved far enough to open the steam-port leading to the main cylinder, since the possible travel of G is greater than that of D.

The supplemental piston B cannot strike the heads of the smaller cylinder, for in its alternate passage beyond the exhaust-ports X and X_1 it cushions on the vapor entrapped in the ends of the cylinder.

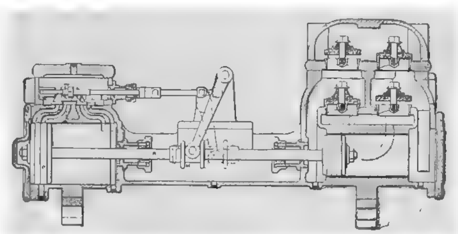


FIG. 100.—Dean duplex pump.

The Dean duplex pump, Fig. 100, varies but very little in its valve-gear from the Worthington and Knowles pumps of the duplex model, while its water-cylinder and valves are the same as the Knowles pattern.

These sectional views represent the valve-gear and general action of a majority of the boiler feed-pumps on the market, and a further illustration may not be desirable.

The basket-strainer, Fig. 101, is a most desirable appendage at or near the entering end of a suction-pipe drawing water from a river or pond, and consists of a perforated plate or frame covered with wire cloth, slid into a cylinder, as shown, with a cover and yoke which allow of cleaning and of removal of fish and floating vegetation (eels often give much trouble in suction-pipes) without having to take up a submerged strainer.

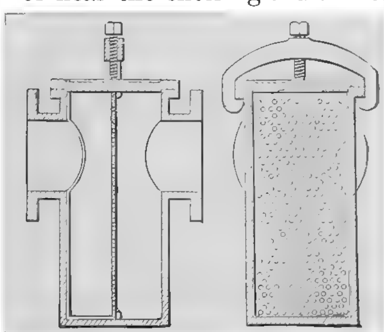


FIG. 101.—Basket-strainer.

Pumps should go slow for their best work, especially when drawing from long suction-pipes; although a large air-chamber on the suction-pipe near the pump will help matters in regard to the pounding caused

by the piston by rapid motion in running away from the water-supply.

In the pumping of hot water this effect is so strong that at temperatures near to the boiling-point a pump will not lift the water; in such cases the pump should be set below the bottom of the hot-water tank.

The air-chamber on the discharge side of a pump performs a most important service in equalizing the flow of water through long pipes and for preventing the noise and hammering in the pipe-lines by the elasticity of the air in the chamber which compensates for the intermittent action of the piston.

The volume of the air-chamber varies in different makes of pumps from 2 to $3\frac{1}{2}$ times the volume of the water-cylinder in single-cylinder pumps, and from 1 to $2\frac{1}{2}$ times the volume of the water-cylinder in the duplex type. The volume of the water-cylinder is represented by the area of the water-piston multiplied by the length of stroke.

For single-cylinder, boiler-feed pumps and those employed for elevator and similar service the volume of the air-chamber should be 3 times the volume of the water-cylinder, and for duplex pumps not less than twice the volume of the water-cylinder. High-speed pumps, such as fire-pumps, should be provided with air-chambers containing from 5 to 6 times the volume of the water-cylinder.

The diameter of the neck should not exceed one-third the diameter of the chamber. When the pumps work under pressure exceeding 85 or 90 pounds per square inch, it is frequently found that the air gradually disappears from the air-chamber, the air passing off with the water by absorption. In this case air should be supplied to the air-chamber unless the pump runs at very low speeds, say, from 10 to 20 strokes for the smaller sizes and from 3 to 5 strokes per minute for pumping-engines. At higher speed and with no air in the air-chamber the valves are apt to seat heavily and cause more or less jar and noise, and the flow of water will not be uniform. In large pumping-plants small air-pumps are employed for keeping the air-chambers properly charged. In smaller plants an ordinary bicycle-pump and a piece of rubber tubing are used to good advantage. The water-level in the air-chamber should be kept down to from one-fourth to one-third the height of the air-chamber for smooth running at medium and high speeds.

CHAPTER VIII

INCRUSTATION IN BOILERS, AND ITS REMEDY

APART from the frequent blowing off of boilers for discharging the floating material that otherwise would settle upon the tubes or plates and form incrusting scale, there are ingredients needed to so change the chemical combination of the scale-forming matter that it may be made soluble at the boiler temperature and blown out or changed into solid particles that do not crystallize on the surface of tubes and plates, and that can be partially blown out or cleaned out at stated periods.

A knowledge of the nature of the scale-forming material in water that is to be used for steam-making is essential, and if this cannot be readily obtained from tests, a sample of the scale may give a clew to its chemical composition. The principal compounds found in such material are carbonate of lime, sulphate of lime, carbonate of magnesia, and sulphate of magnesia; any of which may be nearly pure, or combined or mixed with clay, fine sand, or mud, which tends to modify the hardness of the scale or settles in the boiler as a sludge. The scale from water containing carbonate of lime alone is not as hard as the scale from the sulphate, and is detached much easier. The sulphate scale may be recognized by its sulphurous fumes when heated.

The base of many boiler-compounds made for feeding to boilers with the water is carbonate of soda. Caustic soda and sodium tannate, or extract of oak, sumac, and hemlock bark—mixed with sal soda, sal ammoniac, and triphosphate of sodium—are also used. For the sulphate-of-lime water caustic soda gives a strong reaction, in which sulphate of soda is formed—which is soluble—and hydrate of lime falls as a powder.

PURIFICATION OF BOILER FEED-WATER

In large steam-plants the purification of the water before feeding it to the boilers is most desirable in the line of economy and durability. For this purpose a water-purifying apparatus is in order, and

we illustrate in Fig. 102 an automatic one in use by the Chicago & Northwestern Railway Co. for purging the water for their locomotive service. The following table of the causes of incrustation and corrosion, with their effect and remedies has been formulated to meet these troubles with approved treatment:

TABLE XVII.—CAUSES OF INCRUSTATION, CORROSION, AND THEIR REMEDIES.

Cause of trouble.	Incrustation.	Treatment of water.
Carbonate of lime..............	Soft scale......	Slaked lime, sal-soda.
" of magnesia..........	"	" " "
Sulphate of lime...............	Hard scale	Sal-soda, caustic soda.
" of magnesia...........	"	Slaked lime and sal-soda.
Chloride of magnesia...........	Corrosion	Sal-soda, or caustic soda.
Sediment of sand, clay, and mud	Precipitation, or soft scale.	Alum, and filter.
Organic matter..............	Foaming and corrosion....	Slaked lime, sal-soda, or caustic soda.
Alkaline water................	Foaming	Frequent blowing off from boiler, or neutralize with hydrochloric acid.
Acid waters....................	Corrosion	Slaked lime, sal-soda.

Triphosphate of sodium may be also used instead of lime, but is somewhat more expensive than the lime treatment.

In the use of a purifying apparatus it is necessary to find by trial how much of the chemicals is required in a saturated mixture with water, which should be stored in a tank from which the proper quantity may be automatically drawn and mixed with the water-supply and allowed to settle in large tanks.

In Fig. 102 is shown a cross-section of the apparatus, which consists of a receiving-tank for the chemicals, *a*, with a filter-screen at *b*, from which the chemicals are drawn into a stirring-tank to keep the mixture uniform, whence they are forced to flow to and mix with the boiler-feed water at a uniform rate by measurement in a tilting-tank. By opening the valve *e* this solution is allowed to run into the chemical tank *d*. To thoroughly mix and keep the solution stirred up in the chemical tank *d*, stirring-blades are fixed on the vertical shaft *g*, which rotates in the centre of this tank. In order to measure and deliver predetermined quantities of the chemical solution, the chemical tank *d* is provided with two pumps, *k* and k^1, Fig. 103, connected at the lower portions to the chemical tank *d* through the T's *l* and l^1.

The upper portions of these pumps have discharge-pipes, m and m^1, which discharge into a funnel, n, attached to an elbow terminating on the hard-water supply-pipe, so that just before the hard water passes out of this pipe the chemical solution is mixed with it.

To obtain the best results it is essential that the quantity of the standard chemical solution and hard water be mixed in proper proportions, and also that this be done regularly whenever the apparatus is being used; also that it be done economically. To do

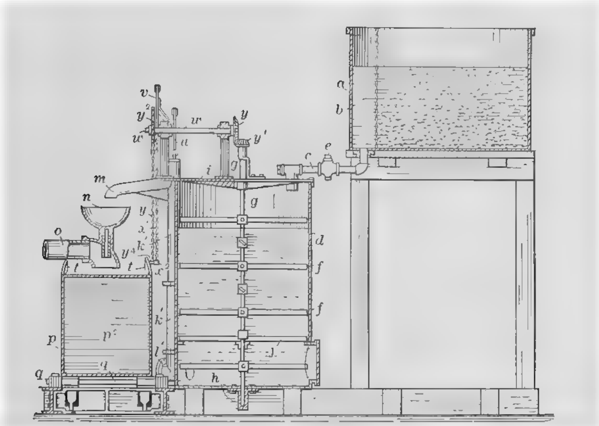


FIG. 102.—Cross-sections of purifying apparatus.

this a tilting-vessel, p, Fig. 103, is used. It is supported on a shaft, q, which is located directly under the elbow from which the mixed hard water and chemical solution are discharged.

This tilting- and measuring-vessel is divided into two compartments of equal capacity, p^1 and p^2. When it is in the position shown in Fig. 103, the mixture of hard water and chemicals falls from the discharge-elbow o into the compartment p^1. When this compartment is nearly filled it counterbalances the weight of the other compartment, p^2, so that the vessel tilts until it strikes the spring 30, emptying the contents of the compartment p^1, and at the same time bringing the other compartment, p^2, under the discharge-elbow o. When this in turn is filled it reverses the movement of the tilting-vessel p, emptying the contents of the compartment p^2, and bringing the compartment p^1 again under the elbow o. For convenience these compartments, p^1 and p^2, are made of such size that 100 gallons of

water are required to fill them to the point where they commence to tilt and empty their contents.

Having determined the amount of a standard solution of chemicals required to precipitate the scale-forming compounds from, say, 100 gallons of any hard water, it is necessary to mix it with the 100 gallons of hard water in one of the compartments p^1 or p^2. This is

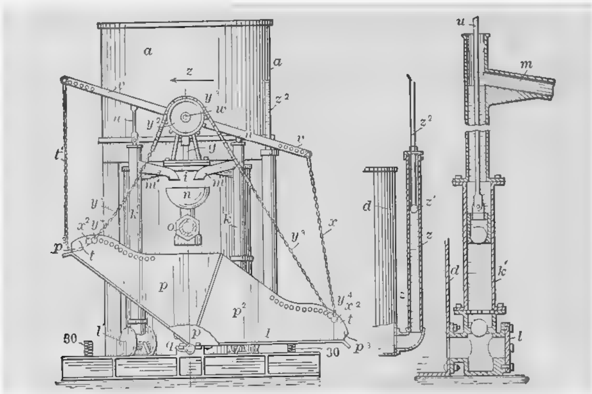


FIG. 103.—Elevation-tanks and pump.

done by regulating the length of the stroke of the pumps k and k^1, which pump the standard chemical solution from the tank d into the funnel n. These pumps, k and k^1, are operated by the tilting-vessel p in the following manner:

The plungers u are connected to a walking-beam, v, which is rotably mounted on the shaft w. The ends of this walking-beam are connected, by means of the chains x and x^1, with studs, x^2, on each end of the tilting-vessel. If the parts are in the position shown in Fig. 103, when the tilting-vessel p is tilted downwardly to the left the plunger of the pump k is raised so that a quantity of the standard chemical solution is delivered into the funnel n, and flows with the hard water into the compartment p^2. When 100 gallons are in it, the tilting-vessel p operates in the opposite direction, causing the other pump, k^1, to operate, and delivers a quantity of the standard chemical solution into the funnel n, whence it flows with the hard water into the compartment p^1. It will be understood that the hard water is running constantly through the elbow o, and that the two pumps k and k^1 are intermittent in their action. The quantity of the standard

chemical solution delivered at each stroke of these pumps is regulated by the length of the strokes. This can be adjusted by the length of the chains x and x^1, so that a predetermined quantity of chemical solution will be delivered at each stroke. From this description it will readily be seen that a fixed quantity of chemical solution is discharged into the elbow o and flows with the hard water into each compartment of the tilting-vessel p, in proportion to the amount of hard water that is required to cause this vessel to tilt.

It is desirable to automatically and economically operate the vertical shaft g in the chemical tank d, so that the horizontal blades attached to it will keep the chemical mixture thoroughly agitated. To do this it is geared to the horizontal shaft w by the pinions y and y^1. The other end of the horizontal shaft w is provided with a sprocket-wheel, y^2, around which a link-belt chain passes, the ends of this chain being attached to the ends of the tilting-vessel by the studs y^4. It

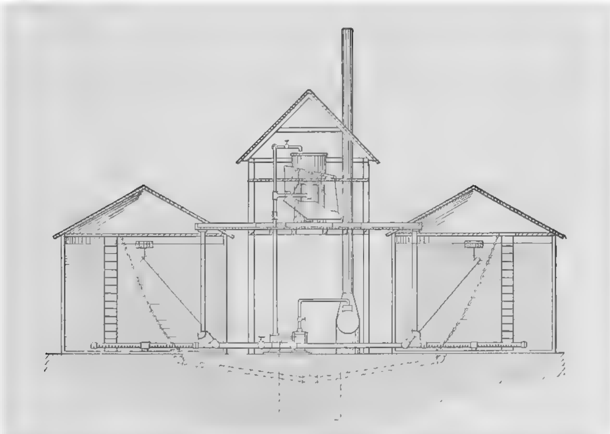


Fig. 104.—Pump-house and settling-tanks.

will readily be seen by this arrangement that whenever the tilting-vessel p moves, the stirring-blades attached to the vertical shaft g in the chemical tanks also move, thus agitating the chemical mixture in the tank d.

For convenience in measuring the height in the tank of this chemical mixture, a pipe, z, Fig. 103, is attached to the side of the tank d near its bottom. In this pipe is a float, z^1, attached to a graduated scale, z^2, from which can be read the quantity of liquid in the tank d.

The above-described apparatus automatically mixes the proper quantity of the chemical solution with each 100 gallons of hard water delivered by the steam-pump, and utilizes the weight of the water to furnish power to operate it. The result of this mixture is that the scale-forming matter that was in solution in the hard water is thrown out of solution, but remains in suspension in the treated water. This is separated from the treated water in the following manner:

By referring to Fig. 104 it will be seen that the apparatus is located in the second story of the pump-house, and that the pump-house is located between two tanks placed on the ground. The tilting-vessel above described empties its contents into a wooden box which is provided with troughs leading to the two settling-tanks. These troughs are provided with shut-off gates, so that the water can be run into whichever tank is desired. It will be seen that the troughs empty their contents into vertical pipes that extend to the bottom of the tanks and terminate in elbows, so as not to disturb the clear water drawn from the top.

The water for boiler use is drawn from the float-nozles at the surface of the water, which swing downward as the water-level is drawn down. The tanks are cleaned alternately.

From records of many trials of the effect of incrustation on fuel-consumption in Europe and the United States, it has been found that there is an average loss of 15 per cent. in full by $\frac{1}{16}$-inch scale, and a greater loss as the scale thickens.

THE FACTOR OF EVAPORATION

To determine the efficiency of a boiler, or the amount of water evaporated by a pound of fuel, it is necessary to reduce the amount of evaporation which actually takes place *from* the temperature of the feed-water at the temperature of the steam, to an equivalent amount at and from 212° F. The factor of evaporation at 212° F. and atmospheric pressure $=1.00$.

Then from the total heat-units in Column 6 of Table XX of the properties of saturated steam for any absolute pressure, subtract the heat-units in the feed-water from 32° F. to its temperature; divide the remainder by the constant 966.1 (the latent heat of steam at 212°

TABLE XVIII.—FACTORS OF EVAPORATION.

Temperature of feed-water, Fahrenheit.	Absolute Pressure of Steam in Pounds per Square Inch.																		
	20.	30.	40.	50.	60.	70.	80.	90.	100.	110.	120.	130.	140.	150.	160.	170.	180.	190.	200.
Degrees.																			
32	1.192	1.199	1.204	1.209	1.212	1.216	1.218	1.221	1.223	1.226	1.228	1.230	1.231	1.233	1.235	1.236	1.238	1.239	1.240
35	1.189	1.196	1.201	1.206	1.209	1.213	1.215	1.218	1.220	1.223	1.225	1.227	1.228	1.230	1.232	1.233	1.235	1.236	1.237
40	1.184	1.191	1.196	1.201	1.204	1.208	1.219	1.213	1.215	1.218	1.220	1.222	1.223	1.225	1.227	1.228	1.230	1.231	1.232
45	1.178	1.185	1.190	1.195	1.198	1.202	1.204	1.207	1.209	1.212	1.214	1.216	1.217	1.219	1.221	1.222	1.224	1.225	1.226
50	1.173	1.180	1.185	1.190	1.193	1.197	1.199	1.202	1.204	1.207	1.209	1.211	1.212	1.214	1.216	1.217	1.219	1.220	1.221
55	1.168	1.175	1.180	1.185	1.188	1.192	1.194	1.197	1.199	1.202	1.204	1.206	1.207	1.209	1.211	1.212	1.214	1.215	1.216
60	1.163	1.170	1.175	1.180	1.183	1.187	1.189	1.192	1.194	1.197	1.199	1.201	1.202	1.204	1.206	1.207	1.209	1.210	1.211
65	1.158	1.165	1.170	1.175	1.178	1.182	1.184	1.187	1.189	1.192	1.194	1.196	1.197	1.199	1.201	1.202	1.204	1.205	1.206
70	1.153	1.160	1.165	1.170	1.173	1.177	1.179	1.182	1.184	1.187	1.189	1.191	1.192	1.194	1.196	1.197	1.199	1.200	1.201
75	1.148	1.155	1.160	1.165	1.168	1.172	1.174	1.177	1.179	1.182	1.184	1.186	1.187	1.189	1.191	1.192	1.194	1.195	1.196
80	1.142	1.149	1.154	1.159	1.162	1.166	1.168	1.171	1.173	1.176	1.178	1.180	1.181	1.183	1.185	1.186	1.188	1.189	1.190
85	1.137	1.144	1.149	1.154	1.157	1.161	1.163	1.166	1.168	1.171	1.173	1.175	1.176	1.178	1.180	1.181	1.183	1.184	1.185
90	1.132	1.139	1.144	1.149	1.152	1.156	1.158	1.161	1.163	1.166	1.168	1.170	1.171	1.173	1.175	1.176	1.178	1.179	1.180
95	1.127	1.134	1.139	1.144	1.147	1.151	1.153	1.156	1.158	1.161	1.163	1.165	1.166	1.168	1.170	1.171	1.173	1.174	1.175
100	1.122	1.129	1.134	1.139	1.142	1.146	1.148	1.151	1.153	1.156	1.158	1.160	1.161	1.163	1.165	1.166	1.168	1.169	1.170
105	1.116	1.123	1.128	1.133	1.136	1.140	1.142	1.145	1.147	1.150	1.152	1.154	1.155	1.157	1.159	1.160	1.162	1.163	1.164
110	1.111	1.118	1.123	1.128	1.131	1.135	1.137	1.140	1.142	1.145	1.147	1.149	1.150	1.152	1.154	1.155	1.157	1.158	1.159
115	1.106	1.113	1.118	1.123	1.126	1.130	1.132	1.135	1.137	1.140	1.142	1.144	1.145	1.147	1.149	1.150	1.152	1.153	1.154
120	1.101	1.108	1.113	1.118	1.121	1.125	1.127	1.130	1.132	1.135	1.137	1.139	1.140	1.142	1.144	1.145	1.147	1.148	1.149
125	1.096	1.103	1.108	1.113	1.116	1.120	1.122	1.125	1.127	1.130	1.132	1.134	1.135	1.137	1.139	1.140	1.142	1.143	1.144
130	1.090	1.097	1.102	1.107	1.110	1.114	1.116	1.119	1.121	1.124	1.126	1.128	1.129	1.131	1.133	1.134	1.136	1.137	1.138
135	1.085	1.092	1.097	1.102	1.105	1.109	1.111	1.114	1.116	1.119	1.121	1.123	1.124	1.126	1.128	1.129	1.131	1.132	1.133
140	1.080	1.087	1.092	1.097	1.100	1.104	1.106	1.109	1.111	1.114	1.116	1.118	1.119	1.121	1.123	1.124	1.126	1.127	1.128
145	1.075	1.082	1.087	1.092	1.095	1.099	1.101	1.104	1.106	1.109	1.111	1.113	1.114	1.116	1.118	1.119	1.121	1.122	1.123
150	1.070	1.077	1.082	1.087	1.090	1.094	1.096	1.099	1.101	1.104	1.106	1.108	1.109	1.111	1.113	1.114	1.116	1.117	1.118
155	1.064	1.071	1.076	1.081	1.084	1.088	1.090	1.094	1.095	1.098	1.100	1.102	1.103	1.105	1.107	1.108	1.110	1.111	1.112
160	1.059	1.066	1.071	1.076	1.079	1.083	1.085	1.088	1.090	1.093	1.095	1.097	1.098	1.100	1.102	1.103	1.105	1.106	1.107
165	1.054	1.061	1.066	1.071	1.074	1.078	1.080	1.083	1.085	1.088	1.090	1.092	1.093	1.095	1.097	1.098	1.100	1.101	1.102
170	1.049	1.056	1.061	1.066	1.069	1.073	1.075	1.078	1.080	1.083	1.085	1.087	1.088	1.090	1.092	1.093	1.095	1.096	1.097
175	1.044	1.051	1.056	1.061	1.064	1.068	1.070	1.073	1.075	1.078	1.080	1.082	1.083	1.085	1.087	1.088	1.090	1.091	1.092
180	1.038	1.045	1.050	1.055	1.058	1.062	1.064	1.067	1.069	1.072	1.074	1.076	1.077	1.079	1.081	1.082	1.084	1.085	1.086
185	1.033	1.040	1.045	1.050	1.053	1.057	1.059	1.062	1.064	1.067	1.069	1.071	1.073	1.074	1.076	1.077	1.079	1.080	1.081
190	1.028	1.035	1.040	1.045	1.048	1.052	1.054	1.057	1.059	1.062	1.064	1.066	1.067	1.069	1.071	1.072	1.074	1.075	1.076
195	1.023	1.030	1.035	1.040	1.043	1.047	1.049	1.052	1.054	1.057	1.059	1.061	1.062	1.064	1.066	1.066	1.069	1.070	1.071
200	1.018	1.025	1.030	1.035	1.038	1.042	1.044	1.047	1.049	1.052	1.054	1.056	1.057	1.059	1.061	1.062	1.064	1.065	1.066
205	1.012	1.019	1.024	1.029	1.032	1.036	1.038	1.041	1.043	1.046	1.048	1.050	1.051	1.053	1.055	1.056	1.058	1.059	1.060
210	1.007	1.014	1.019	1.024	1.027	1.031	1.033	1.036	1.038	1.041	1.043	1.045	1.046	1.048	1.050	1.051	1.053	1.054	1.055
212	1.005	1.012	1.017	1.022	1.025	1.029	1.031	1.034	1.036	1.039	1.041	1.043	1.044	1.046	1.048	1.049	1.051	1.052	1.053

F.), and the quotient will give the factor of evaporation as per Table XVIII. For example: the factor of evaporation for 100 pounds absolute pressure—85.3 gauge-pressure—from water at 212° F. and feed-water at 100° F., will be $\frac{1181.9-68}{966.1}=1.153$, as in the table, Column 100, and opposite 100° F. in the first column.

The total heat-units from the feed-water temperature in the steam, at any given pressure, may be readily obtained by multiplying the total heat-units at 212° F. as a constant (966.1), by the factor of evaporation for the feed-water temperature in the first column at the intersection of its line in the columns of absolute pressure, Table XVIII.

Intermediate temperatures and pressures may be obtained by interpolation. The use of the factor of evaporation is apparent as a ready method for obtaining the actual number of pounds of water evaporated in a boiler from and at a temperature of 212° F. per pound of coal or of combustible, if the combustible value of the coal is known.

For example: with feed-water at 100° F., average steam-pressure during trial 75.3 gauge = 90 pounds absolute, with say 40 pounds coal burned for any unit of time and 340 pounds of water fed to the boiler for the same unit of time; then $\frac{340}{40}=8.5$ pounds of water evaporated at 100° F. per pound of coal. Then for the evaporation from and at 212° F., the factor of evaporation for 100° and 90 pounds is by the table 1.151, and $8.5 \times 1.151 = 9.27$ pounds water evaporated at 212° F.

THE JET-CONDENSER

Where a sufficient quantity of water suitable for boiler-feeding purposes is available, the jet-condenser, being the simplest and easiest to operate, is preferable. Where, however, water suitable for boiler-feeding is not available, a surface-condenser may be used. In this type the steam is condensed in a condensing-chamber on the surface of tubes through which cold water is circulating, and the distilled water so furnished may be again fed to the boilers. Where any considerable amount of cylinder-oil is used, some provision must be made with surface-condensers to remove this oil before the water is fed to the boilers. With either type the quantity of water to be

circulated through the condenser should be from twenty to forty times the quantity of steam to be condensed, depending upon the temperature of the water available for condensing purposes.

Condenser manufacturers have recently introduced several types of self-cooling condensers by which the hot water delivered from the condenser-pumps can be cooled and reused, so that with water sufficient in quantity for boiler-feed purposes only, the plant may be located at any convenient point and still retain the fuel-saving and other benefits of high steam-pressures and condensers.

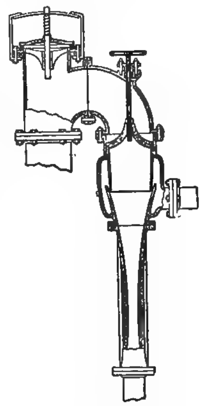


Fig. 105.—Siphon condenser.

The condenser-head, shown in section in Fig. 105, consists of a suitable steam-chamber, usually in the form of a large return-bend. This is fitted with a relief-valve at the top which closes automatically, due to its own weight aided by a light spring. When a vacuum exists in the condenser-head, the valve is pressed more firmly against its seat by atmospheric pressure of about 15 pounds per square inch.

Connected with one of the openings in the steam-chamber, or return-bend, is the regulating-nozle, which is movable vertically and is raised by means of a threaded stem and hand-wheel. The nozle regulates the width of the inlet-orifice for the condensing water according to the load, the water entering the side of the nozle-chamber and surrounding the nozle, flows in a thin sheet or film through the annular orifice formed between the nozle and its seat.

Below the chamber is connected the throat or combining-tube, the bore of which gradually contracts toward the middle of its length and then enlarges toward the lower end, where it is connected to the tail-pipe, which extends to 34 feet below the nozle and dips into the hot-well for the purpose of a water-seal to prevent air entering the pipe and to resist the atmospheric pressure from without. Therefore, if the tail-pipe were less than 34 feet long, measured between the points mentioned, water supplied to the condenser would not leave the latter without the use of a pump. But with a fall of 34 feet, a given quantity of water admitted around the hollow cone, or nozle, causes the discharge of a corresponding amount into the hot-

well, so that the level in the condenser never can rise to the water-inlet.

Water passing through the annular orifice formed by the hollow cone flows downward in a cone-shaped film into the contracted throat, where its velocity is sufficiently increased to enable it to carry air along with it, thus producing a vacuum in the exhaust-pipe. Steam

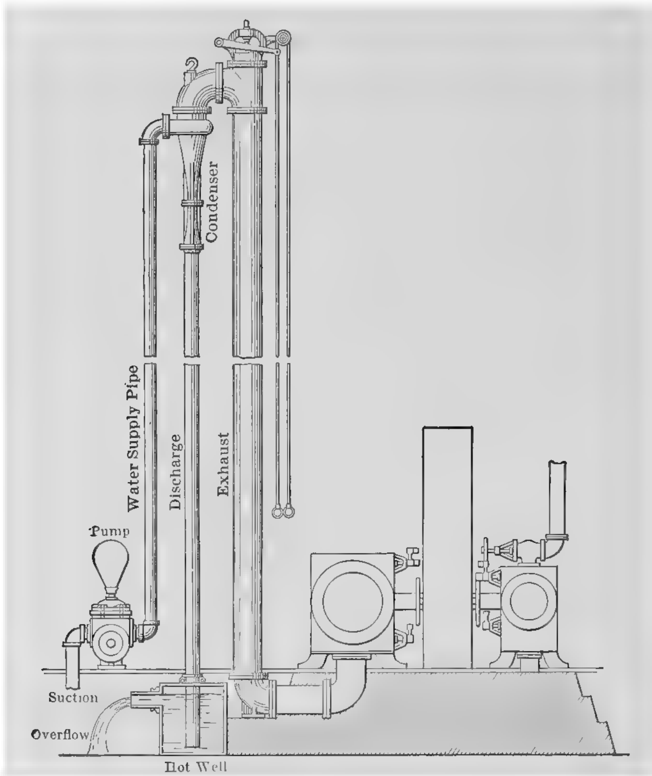


FIG. 106.—Siphon condenser connected.

flows downward through the regulating-nozle and into the cone-shape film of water, where it is condensed. The continuous condensation of steam and the ability to get rid of the water may sometimes cease, when the exhaust-valve will be raised, allowing the steam to escape into the atmosphere. If it becomes necessary to break the vacuum, the relief-valve can be opened from the engine-room floor by means of chains connected with a lever attached to the valve. There are a

number of this type of condensers, of various models, on the market, all involving the same principles as here shown.

The ejector-condenser, Fig. 107, is of the Korting type, with a three-way valve by which the exhaust-steam is passed to the atmosphere or is condensed by the multiple-nozle water-jet. The high velocity of the water-jet past the angular orifices in the nozle maintains the required vacuum without recourse to a pump or long vertical pipe.

A sectional view of a Worthington direct-acting jet-condenser is given in Fig. 108. In all essential features it is a duplex, direct-acting

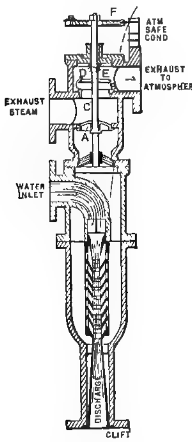


Fig. 107.—Ejector-condenser.

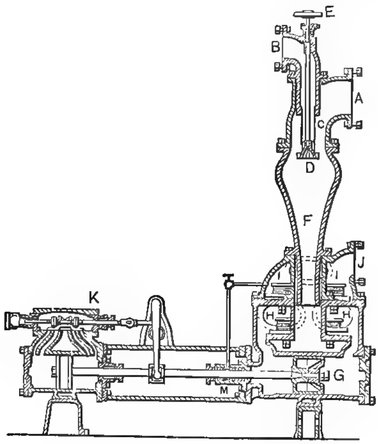


Fig. 108.—Jet-condenser pump.

pump with a condensing-chamber or cone connected with the pump-suction or suction-chamber. The exhaust-pipe from the engine is connected at A, and the pipe supplying the injection-water is connected at B, which point on the condenser should never be more than 18 feet above the surface of the water from which the supply of condensing water is to be drawn. The discharge from the condenser flows out at J through a pipe leading to the hot-well.

When the pump is started, a partial vacuum is created in the suction-chamber above the valves H, H, in the cone F, in the exhaust-pipe at A, and in the injection-pipe connected at B. As soon as sufficient air has been exhausted, water begins to flow through the pipe C

and the spray-nozle D. Continued motion of the pump increases the vacuum up to the working point, 26 or 27 inches. Water issuing from the nozle D is broken into a fine spray which completely fills the condensing-chamber or cone beneath it, so that upon starting the engine the exhaust-steam is compelled to flow into the spray of cold water.

The mixture of condensed steam and injection flows downward through the tapered throat F into the suction-chamber of the pump with sufficient velocity to carry with it any air that may have leaked into the exhaust-pipe, together with the air brought in with the injection-water. The direction of the water in the pump may be easily traced, the pump discharging both water and air through the discharge-valves I, I and outlet J.

Many manufacturers of pumps are now making the condenser attachments to their vacuum-pumps.

WATER REQUIRED FOR CONDENSING THE EXHAUST

Evidently the heat given up by the steam must equal the heat gained by the cooling water, and for each pound of steam condensed there will be a certain number of pounds of cooling water used under a given set of conditions. This makes it possible to determine the theoretical ratio between the weight of condensed steam and the weight of cooling water used, and this theoretical ratio will for the jet-condenser correspond to the actual ratio. For the surface-condenser the amount of cooling water used will be about 20 per cent. in excess of the theoretical value.

The heat removed from a pound of steam, with variable terminal pressure, is but slight within practical limits. For instance, at 30 pounds absolute terminal pressure, the heat contained is 1,190.3 thermal units, while at 5 pounds absolute pressure it is 1,163.5 thermal units, a difference of a little over 2 per cent.; hence, it is not necessary to figure on small differences in terminal pressure. Table XIX shows the ratio of the cooling water to the condensed steam, or, in other words, the number of pounds of cooling water needed per pound of steam for the terminal pressure of 15 pounds absolute, and for final temperatures of the condensed steam from 90° to 134° F.

TABLE XIX.—POUNDS OF WATER REQUIRED TO CONDENSE 1 POUND OF STEAM AT EXHAUST-PRESSURE OF 15 POUNDS ABSOLUTE, IN JET-CONDENSERS.

Temperature air-pump discharge.	ENTERING TEMPERATURE OF INJECTION-WATER.											
	35	40	45	50	55	60	65	70	75	80	85	90
	Pounds of condensing water required per pound of steam.											
90	20.0	22.0	24.4	27.5	31.4	36.7	44.0	55.0	73.3	110.0	220.0	.....
92	19.2	21.1	23.4	26.1	29.7	34.3	40.7	49.9	64.6	91.5	156.8	549.0
94	18.6	20.3	22.4	24.9	28.1	32.2	37.8	45.7	57.7	78.1	121.8	274.0
96	17.9	19.5	21.4	23.6	26.7	30.4	35.3	42.1	52.1	68.4	99.4	182.3
98	17.3	18.8	20.6	22.7	25.4	28.7	33.1	39.0	47.5	60.7	84.0	136.5
100	16.8	18.2	19.8	21.1	24.2	27.2	31.1	36.3	43.6	54.5	72.7	109.0
102	16.2	17.5	19.1	20.9	23.1	25.9	29.4	34.0	40.3	49.5	64.0	90.7
104	15.7	17.0	18.4	20.1	22.2	24.7	27.8	31.9	37.4	45.2	57.2	77.6
106	15.3	16.4	17.8	19.4	21.3	23.6	26.4	30.1	35.0	41.7	51.6	67.7
108	14.8	15.9	17.2	18.7	20.4	22.5	25.2	28.5	32.8	38.6	47.0	60.1
110	14.4	15.4	16.6	18.0	19.6	21.6	24.0	27.0	30.9	36.0	43.2	54.0
112	14.0	15.0	16.1	17.4	18.9	20.7	22.9	25.7	29.1	33.6	39.9	49.0
114	13.6	14.5	15.6	16.8	18.2	19.9	22.0	24.5	27.6	31.6	37.1	44.8
116	13.3	14.1	15.1	16.3	17.6	19.2	21.1	23.3	26.2	29.8	34.6	41.3
118	12.9	13.7	14.7	15.8	17.0	18.5	20.2	22.3	24.9	28.2	32.5	38.3
120	12.6	13.4	14.3	15.3	16.5	17.8	19.5	21.4	23.8	26.7	30.6	35.7
122	12.3	13.0	13.9	14.8	15.9	17.2	18.7	20.5	22.7	25.4	28.9	33.4
124	12.0	12.7	13.5	14.4	15.4	16.7	18.1	19.7	21.8	24.2	27.3	31.4
126	11.7	12.4	13.1	14.0	15.0	16.1	17.4	19.0	20.9	23.1	26.0	29.6
128	11.4	12.1	12.8	13.6	14.5	15.6	16.9	18.3	20.0	22.1	24.7	27.9
130	11.2	11.8	12.5	13.2	14.1	15.1	16.3	17.7	19.3	21.2	23.6	26.5
132	10.9	11.5	12.2	12.9	13.7	14.7	15.7	17.1	18.6	20.3	22.5	25.2
134	10.7	11.2	11.9	12.6	13.4	14.3	15.3	16.5	17.9	19.6	21.6	24.0

This table has been computed from the formula $Q = \frac{1{,}190 - T}{T - t}$, in which Q = quantity of water in pounds to condense 1 pound of steam, or 25.85 cubic feet at exhaust temperature of 213.1° F.; T = temperature of water discharged from the condenser; t = the difference in temperature between the injection- and the discharge-water, and 1,190 = the total heat of the steam, plus the loss from radiation in the operation of the condenser.

The surface-condenser, so useful in the line of economy where the cost of water claims a saving of its waste in the jet-condenser, comes to the front in connection with the cooling-tower as an economizer in the generation of steam-power. There are a number of models in design with claims of efficiency.

In Fig. 109 is shown a two-section condenser with a cast-iron shell and brass tubes. The difference in expansion between the brass

tubes and the cast-iron shell is provided for by stuffing-boxes and glands at one or both ends. The cooling water enters through the

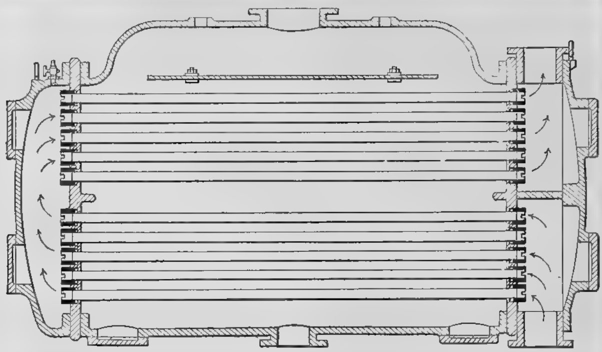


FIG. 109.—Surface-condenser.

lower tier and discharges through the upper one for the best efficiency of the condenser, with the steam passing in the opposite direction.

A three-section surface-condenser and heater is shown in Fig. 110. The upper section is divided in two parts as a feed-water heater, through which part of the water used for condensing is passed through

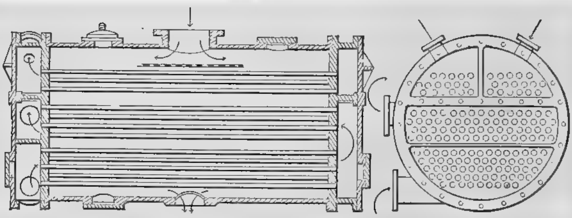


FIG. 110.—Combination condenser.

one part of the upper section and returned by the other side-chamber. By this arrangement the feed-water is heated to as high a temperature as practicable by first contact with the exhaust-steam.

The double-tube type, Fig. 111, is one in which the shell encloses both tube-heads at one end, in one of which the central tubes are carried through the inner compartment, and both inner and outer concentric tubes are expanded in their respective heads.

The outside tubes are capped or welded close at their farther ends. The circulation is made complete by its discharge from the inner

to the outer tube, which is the condensing-surface; thus all troubles from expansion are avoided.

A novel system of surface-condensation is shown in Fig. 112, in which a cylinder is filled with small brass tubes, open for receiving a spray-jet of water at one end; a cone and suction-blower are at the other end, as well as the usual vacuum-pump. In its action a spray-jet of water is thrown against the tubes at one end, and with a large volume of air is drawn through the tubes by a suction-blower at the end of the conical chamber. The water is vaporized, and with the air takes up the heat of the exhaust-steam, which is discharged in a vapor by the blower. The economical claim for this arrangement is that but 1 pound of water is used for condensing 1 pound of steam.

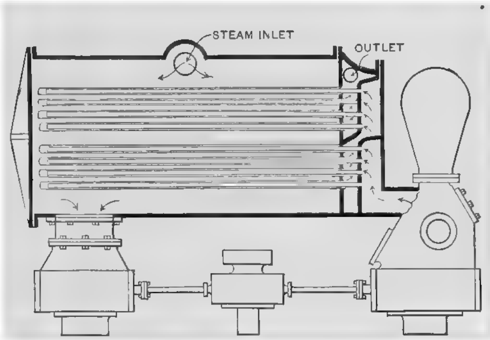


Fig. 111.—Concentric tube-condenser.

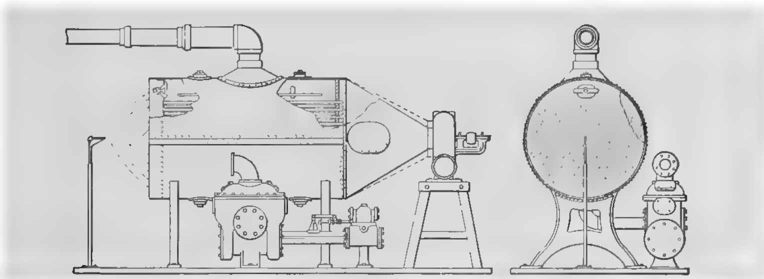


Fig. 112.—Spray surface-condenser.

The large number of oil-extracting devices on the market, of various models, need no discussion as to their merits, as each has a claim to be the best. They are a great need and in general use, and are usually connected in the exhaust-pipe near the surface-condenser. The leading principle of action of these separators is their sudden deflection of the passing steam by an apron, which may be curved or flat, pierced with slots, holes, or with corrugated surfaces, arranged to catch the oil and drain it to a receptacle below.

In Fig. 113 is shown a vertical and a horizontal pattern of the Austin Separator Co., and in Fig. 114 a separator of the Lippincott pattern, with a broad spherical apron at A and catch-plates at B, C, D, the steam being deflected around the outside of the spherical plate A.

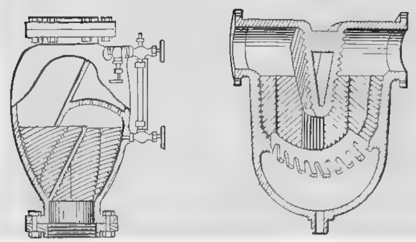


Fig. 113.—Oil-separators.

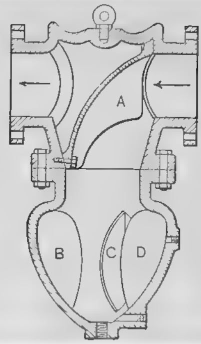


Fig. 114.—Lippincott separator.

A combined vacuum air-pump and water-circulating pump for large condensing-engines is shown in Fig. 115. It is of the Conover type of the Watson Machine Co. Its compact design makes it a good study for the student and engineer. It is operated by a pair of compound Corliss cylinders, with dash-pots complete. The beam-ends connect with the steam-cylinders, and at mid-distance to its centre are connected to the pumps, and from one of these to the crank, the shaft of which is seen in the centre of the illustration. The shaft carries a fly-wheel at the rear end and drives the governor. The air-pump is single-acting; the circulating-pump is a double-acting trunk pattern, located at the right. The receiver is attached to the frame just below the beam. This type of air- and circulating-pumps is made for engines of from 5,000 to 20,000 horse-power.

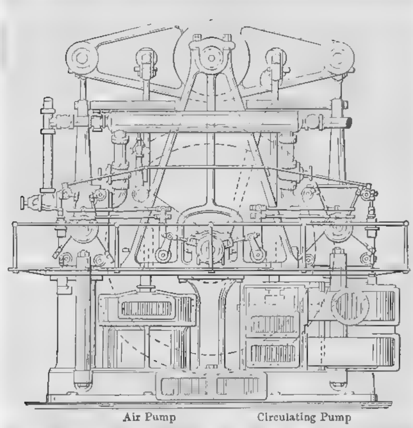


Fig. 115.—Air- and circulating-pump.

A novel air-pump for medium-sized condensing-engines is the Edwards type, Fig. 116, which has no suction-valves. Ports around the cylinder are opened by passing the piston past them to the bottom of the cylinder.

The water and air enter above the piston and are discharged through valves above, which are water-sealed; the discharged water flowing over a dam, as shown by the arrow. A water-filled cup seals the piston-rod below the stuffing-box gland. The descent of the piston forms a vacuum above it, which is a powerful draught at the moment of opening of the cylinder-ports by the piston.

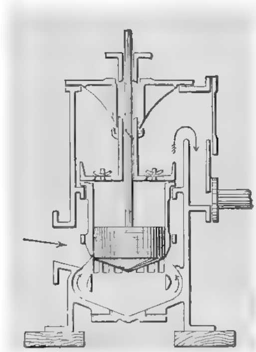


FIG. 116.—Edwards's air-pump.

WATER-COOLING TOWERS

The saving of water in locations where it is deficient for the necessities of steam-power is a matter of great importance, as in arid regions, and of economy, where its cost is of material amount.

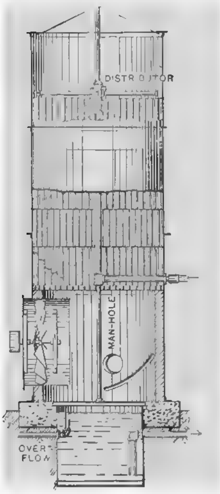


FIG. 117.—Air-cooling tower.

The main feature of the cooling-tower is derived from the intimate contact of cool air, circulated by a fan driven by any convenient power, or by the natural draught caused by the heating of the air by contact of the falling spray or sheets of hot water. In this manner the hot water from a jet- or surface-condenser may be cooled sufficiently for use again in the condenser.

The cooling-towers are filled in a variety of material and forms, such as hanging curtains of galvanized-iron netting, or strips of thin wood and tile in tiers crossing each other, so arranged as to give the greatest wet surface and also the greatest area of airway.

In Fig. 117 we illustrate by a section the Worthington water-cooling tower, which consists of a cylindrical steel shell open at the top, supported upon a suitable foundation, and having fitted at one side a fan, the function of which is to circulate a current of air through

the tower and filling This filling consists of layers of cylindrical tubular tiling, which rest upon a grating supported by a brick wall extending around the circumference of the tower. The heated discharge-water from the condenser enters the tower at the side, passes up the central pipe, and is delivered on the upper layer of tiling and over the whole cross-section of the tower by a distributing device con-

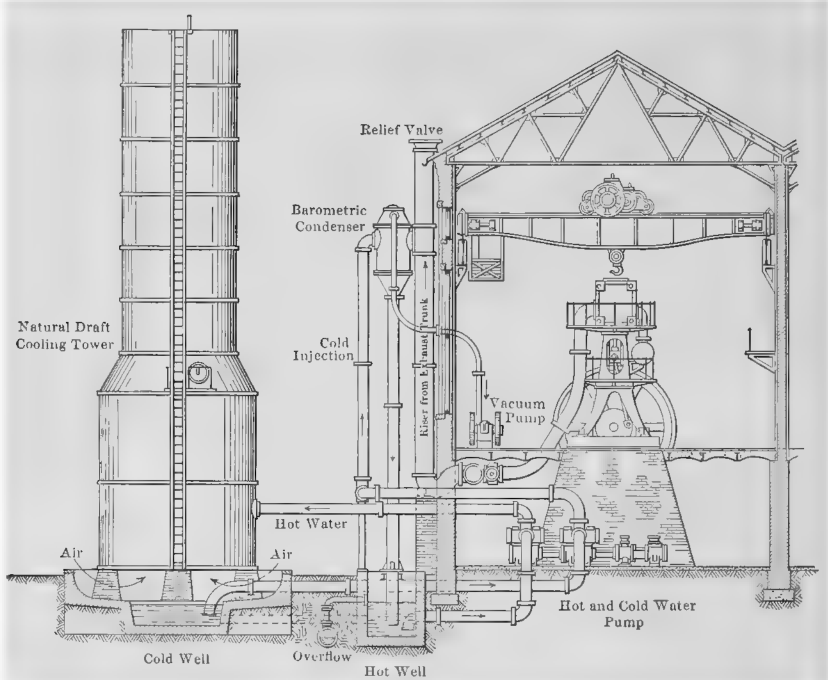


FIG. 118.—High-vacuum installation with cooling-tower.

sisting of four pipes, which are caused to rotate about the central water-pipe by the simple reaction of the jets of heated water issuing from one side of each pipe. The water thus delivered spreads over the outside and inside surfaces of the walls of the tiling and forms a continuous sheet, which is presented to the action of the air.

In Fig. 118 is represented a complete power-plant of the Worthington model with a jet or barometric condenser, natural draught-cooling tower, combined hot- and cold-water pump, and a vacuum-pump.

It will be seen on inspection that the exhaust from one or a series of engines passes into a trunk-pipe from which a rising pipe leads to the head of the ejector-condenser and on to a relief-valve. The cold-water cylinder of the circulating-pump takes its suction from the cold-water well in the tower and discharges into the head of the jet-condenser. The hot-water cylinder of the same pump takes its suction from the hot-well of the condenser and discharges at the top of the tower in fine streams that trickle over the surface of the tiling in contact with the up-flowing air. In order, however, to obtain the highest vacuum without using an abnormal amount of water to carry off the air, a separate dry-vacuum pump is used, as shown in the illustration. The air that is not carried off by the water is taken from the space under the spray-cone in the condenser. By this means it is possible to get as much as 29 inches of vacuum under the most favorable conditions.

The loss of water is minimized in this arrangement to the amount vaporized to the air in the cooling-tower and the leakages.

CHAPTER IX

STEAM ABOVE ATMOSPHERIC PRESSURE

STEAM under pressure and confined, as in a boiler, has a potential energy due to its pressure, which becomes kinetic, a moving force, when following the piston of an engine from the boiler-pressure or by its force of expansion.

Steam, like fluids under pressure, becomes a force by momentum from its pressure and expansive velocity when impinging on the blades of the steam-turbine.

The derivation of its energy, both potential and kinetic, is from heat in its specific and latent forms, which, combined with water, gives it the elastic properties produced in its vapor.

Heat is the basis of energy in nature, in life, and in work of the most important value to our industries; it has a measured value, the heat-unit or British thermal unit, equivalent to the amount received to raise 1 pound of water at the temperature of its greatest density, 39° F., through 1 degree of the Fahrenheit scale.

DIAGRAM OF STEAM-GENERATION

The rise in temperature of water in its frozen state from absolute zero, its absorption of heat in thermal units, its further absorption of heat in melting and rise of temperature to its boiling-point, and its conversion into steam, are graphically shown in the diagram (Fig. 119) in which the vertical scale represents the temperature from absolute zero, and the horizontal scale the heat-units absorbed during the change from ice to steam.

The divergent lines at the right show the thermal-heat difference of steam at constant volume (C_v) and constant pressure (C_p). All the inclined lines should be slightly curved to show the change in specific heat, but it is not readily shown on so small a diagram.

Steam is treated in its work under different conditions, essential to its economical use:

1. As saturated steam; its condition when generated in quiet contact with its water of generation.

2. As wet steam; its condition when by the violent action of its generation from an overworked or foaming boiler it is loaded with minute vesicles of water containing no latent heat and consequently non-expansive in its working economy; although its specific heat at high pressures and temperatures may evolve a minute portion of the water-vesicle into vapor during expansive work.

3. As dry steam; its condition when it contains no vesicular moisture; it may be saturated steam or with initial or expansive superheat.

4. As superheated steam, which is at a temperature above that of the water from which it is generated, either by some peculiar boiler construction, or at a temperature largely increased by passing through a superheater.

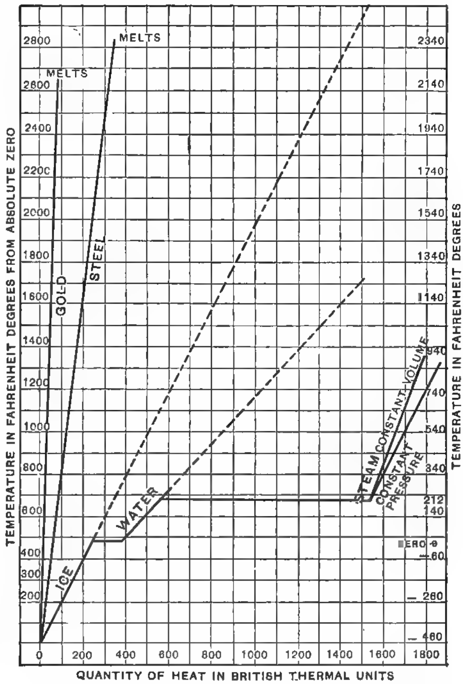


Fig. 119.—Diagram of steam-generation.

The temperature of water and its steam when confined in contact and under pressure may be computed from the sixth root of the absolute pressure in inches of mercury, or the absolute pressure in pounds per square inch multiplied by 2.036. Multiply the sixth root of this product by 176.4, and from the last product subtract 100 for the temperature in Column 3 of the tables of the properties of saturated steam.

For example:

(1) . At 100 pounds absolute, $\sqrt[6]{p \times 2.036} = \sqrt[6]{203.6}$.

As the sixth root is the cube root of the square root of the number, then $\sqrt{203.6} = 14.2688$, and the $\sqrt[3]{14.2688} = 2.42516 \times 176.4 = 427.7 - 100 = 327.6°$, as in Column 3.

The specific heat of water (which has been assigned as 0), at the zero of absolute pressure and 32° F. of temperature, gradually increases in its heat-unit quantity, equivalent to its thermometric temperature, so that at 212° it has absorbed 180.5 heat-units per pound to raise its temperature from 32° to 212° F., with an increasing ratio throughout the range of pressures and temperatures in use. The formula for the specific heat of water as given by Regnault and Rankine for any temperature T, above 32° F., is

(2) . $T - 32° + 0.000{,}000{,}103 \times [(T - 39.1)^3 + (7.1)^3]$ = the number of heat-units imparted to the water as represented in Column 4, Table XX, of the properties of saturated steam.

The latent heat of vaporization of water, Column 5 in the steam table, is from Rankine's formula:

(3) . . $L = 1{,}091.7 - (.698(t - 32°)$, in which t = the thermometric temperature in Column 3.

The total amount of heat in steam, as in Column 6, consists of the amounts in Columns 4 and 5 added, and may be computed from the formula of Regnault, in which

(4) . . $H = 1{,}091.7 + (.305(t - 32°)$, t being the thermometric temperature in Column 3. For example:

$212° - 32° = 180 \times .305 = 54.9$, and $54.9 + 1{,}091.7 = 1{,}146.6$, as in Column 6 of the table of properties of saturated steam. The specific heat of steam is uniform throughout the range of temperatures in contact with its water of generation, and is assigned as 0.305 (water 1).

The specific heat of steam may also be used for obtaining the heat-unit values in Column 6, Table XX. The latent heat of steam at zero pressure is 1,082 heat-units; then the temperature due to the absolute pressure multiplied by the specific heat, plus 1,082, equals the total units above 32° F.

For example: $(212° \times .305) + 1{,}082 = 1{,}146.6$ heat-units, as in Column 6; also at 100.3 gauge-pressure = 115 absolute, the temperature in Column 3: $(337.9 \times .305) + 1{,}082 = 1{,}185.05$, and so on.

Steam has its critical temperature at about 2,052° absolute (1,592° F.), above which the latent heat of evaporation will be zero and there would be no difference between the liquid and vaporous forms, as its liquid volume will have disappeared.

The values in Columns 7, 8, and 9 are relative, so that

$$\text{Column } 7 = \frac{\text{Column } 9}{62.43} \text{ (weight of water per cubic foot);}$$

$$\text{Column } 8 = \frac{1}{\text{Column } 7}, \text{ and Column } 9 = \frac{62.43}{\text{Column } 8}.$$

The volume of steam per pound of water at any temperature may be obtained from the formula:

(5) . . . . $V_2 = V_1 + \dfrac{He}{\dfrac{dp}{dt}\, t}$, in which He = the foot-pound value of the latent heat in Column 5; V_2 = the volume of saturated steam in cubic feet; t = absolute temperature of water and steam; dp = differential pressure per square foot; dt = differential temperature or 1° F.; V_1 = volume of water at temperature t.

The difference in pressure per degree F., at atmospheric pressure for 1 heat-unit, is $\frac{dp}{dt} = \frac{.2945}{1} \times 144 = 42.408$ pounds per square foot; then as water increases in volume .04775 per unit of heat from its maximum density (39.1° F.), its volume per pound is therefore $\frac{1}{62.425} =$ 0.01602 cubic foot. Then $0.01602 \times 1.04775 = 0.01678$, its increased volume for 1 heat-unit.

The cubic feet of steam per pound of water at atmospheric pressure may be computed from the following formula:

(6) . . $0.01678 + \dfrac{966.1 \times 778}{672.66 \times 42.4} = 26.37$, as in Column 7.

For 30 pounds absolute pressure the difference in temperature per pound of pressure from Column 3 is

(7) . . 1.886° F. and $\frac{1}{1.886}=.5302\times 144=76.35$, the ratio.

Then $0.01678+\frac{939\times 778}{710.96\times 76.35}=\frac{730{,}542}{54{,}281.8}=13.458+.017=13.475$, as in Column 7.

For 100 pounds absolute the ratio will be $\frac{1}{.7}=1.43\times 144=205.9$.

(8) . . Then $0.01678+\frac{883.8\times 778}{788.26\times 205.9}=4.227+.017\doteq 4.244$.

Again, for 200 pounds absolute, the ratio of pressure to differential temperature is $\frac{1}{.41}=2.44\times 144=351.3$.

(9) . . Then $0.01678+\frac{843.4\times 778}{842.26\times 351.3}=2.217+0.017=2.234$, the cutting off of fractions making a slight discrepancy from the tables as established.

The total heat-units in Column 6 may be obtained directly from the temperature in Column 3 by the formula

(10) . . 1,091.7+.305 (Column 3)−32, in which 1,091.7 is the total value in heat-units of the vapor of water at the absolute zero of pressure, and .305 the specific heat of saturated steam.

Then, for example, at 100 pounds absolute pressure, from the temperature in Column 3, we have,

(11) . . $327.6-32=295.6\times .305=90.15+1{,}091.7=1{,}181.85$, as in Column 6.

TABLE XX.—PROPERTIES OF SATURATED STEAM—PRESSURES, TEMPERATURE, VOLUME, WEIGHT, ETC.

Barometer-vacuum, inches.	Absolute pressure, pounds.	Temperature of water, Fahrenheit.	Heat-units from 32° F. to temp., Column 3.	Latent heat of vaporization, units.	Total heat, Columns 4 and 5, units.	Volume of 1 pound, cubic feet.	Weight of 1 cubic foot of steam.	Relative volume to water.
1	2	3	4	5	6	7	8	9
2.035	1	102.0°	70.0	1,043.0	1,113.0	330.4	.00303	20,628
4.07	2	126.3	94.4	1,026.1	1,120.5	171.9	.00582	10,730
6.105	3	141.7	109.8	1,015.4	1,125.2	117.8	.00852	7,325
8.14	4	153.1	121.3	1,007.4	1,128.7	89.51	.01117	5,588
10.175	5	162.4	130.6	1,000.9	1,131.5	72.56	.01378	4,530
12.21	6	170.2	138.4	995.4	1,133.8	61.14	.01636	3,816
14.245	7	176.9	145.2	990.7	1,135.9	52.89	.01891	3,302
16.28	8	183.0	151.3	986.5	1,137.8	46.65	.02144	2,912
18.315	9	188.4	156.7	982.7	1,139.4	41.77	.02394	2,607
20.35	10	193.3	161.7	979.3	1,141.0	37.83	.02644	2,361
22.385	11	197.8	166.2	976.1	1,142.3	34.59	.02891	2,151
24.42	12	202.0	170.5	973.1	1,143.6	31.87	.03138	1,990
26.455	13	205.9	174.4	970.4	1,144.8	29.56	.03383	1,845
28.49	14	209.6	178.1	967.8	1,145.9	27.58	.03626	1,721
29.92	14.7	212.0	180.5	966.1	1,146.6	26.37	.03793	1,646
Gauge. .3	15	213.1	181.6	965.3	1,146.9	25.85	.03869	1,614
1.3	16	216.3	184.9	963.0	1,147.9	24.33	.04111	1,519
2.3	17	219.5	188.1	960.8	1,148.9	22.98	.04352	1,434
3.3	18	222.4	191.1	958.8	1,149.9	21.78	.04592	1,359
4.3	19	225.3	193.9	956.7	1,150.6	20.70	.04831	1,292
5.3	20	228.0	196.7	954.8	1,151.5	19.73	.05070	1,281
6.3	21	230.6	199.3	952.9	1,152.2	18.84	.05307	1,176
7.3	22	233.1	201.8	951.2	1,153.0	18.04	.05545	1,126
8.3	23	235.5	204.3	949.5	1,153.8	17.30	.05781	1,080
9.3	24	237.8	206.6	947.8	1,154.4	16.62	.06017	1,038
10.3	25	240.1	208.9	946.3	1,155.2	16.00	.06252	998.4
11.3	26	242.2	211.1	944.8	1,155.9	15.42	.06487	962.3
12.3	27	244.3	213.2	943.2	1,156.4	14.88	.06721	928.8
13.3	28	246.4	215.3	941.8	1,157.1	14.38	.06955	897.6
14.3	29	248.4	217.3	940.4	1,157.7	13.91	.07188	868.5
15.3	30	250.3	219.3	939.0	1,158.3	13.48	.07420	841.3
16.3	31	252.2	221.2	937.7	1,158.9	13.07	.07652	815.8
17.3	32	254.0	223.0	936.4	1,159.4	12.68	.07884	791.8
18.3	33	255.8	224.8	935.1	1,159.9	12.32	.08115	769.2
19.3	34	257.5	226.6	933.9	1,160.5	11.98	.08346	748.0
20.3	35	259.2	228.3	932.7	1,161.0	11.66	.08577	727.9
21.3	36	260.9	230.0	931.5	1,161.5	11.36	.08807	708.8
22.3	37	262.5	231.7	930.4	1,162.1	11.07	.09036	690.8
23.3	38	264.1	233.3	929.3	1,162.6	10.79	.09266	673.7
24.3	39	265.6	234.8	928.1	1,162.9	10.53	.09495	657.5
25.3	40	267.2	236.4	927.0	1,163.4	10.28	.09723	642.0
26.3	41	268.7	237.9	926.0	1,163.9	10.05	.09951	627.3
27.3	42	270.1	239.4	924.9	1,164.3	9.826	.10179	613.3
28.3	43	271.6	240.8	923.9	1,164.7	9.609	.10407	599.9
29.3	44	273.0	242.3	922.9	1,165.2	9.403	.10635	587.0
30.3	45	274.3	243.7	921.9	1,165.6	9.207	.10862	574.7

TABLE XX.—PROPERTIES OF SATURATED STEAM—PRESSURES, TEMPERATURE, VOLUME, WEIGHT, ETC.—(*Continued.*)

Gauge pressure, pounds.	Absolute pressure, pounds.	Temperature of water, Fahrenheit.	Heat-units from 32° F. to temp., Column 3.	Latent heat of vaporization, units.	Total heat, Columns 4 and 5, units.	Volume of 1 pound, cubic feet.	Weight of 1 cubic foot of steam.	Relative volume to water.
1	2	3	4	5	6	7	8	9
31.3	46	275.7°	245.1	920.9	1,166.0	9.018	.11088	563.0
32.3	47	277.0	246.4	920.0	1,166.4	8.838	.11315	551.7
33.3	48	278.3	247.8	919.1	1,166.9	8.665	.11541	540.9
34.3	49	279.6	249.1	918.1	1,167.2	8.498	.11767	530.5
35.3	50	280.9	250.4	917.3	1,167.7	8.338	.11993	520.5
36.3	51	282.2	251.6	916.4	1,168.0	8.185	.12218	510.9
37.3	52	283.4	252.9	915.5	1,168.4	8.037	.12443	501.7
38.3	53	284.6	254.1	914.7	1,168.8	7.894	.12668	492.8
39.3	54	285.8	255.3	913.8	1,169.1	7.756	.12893	484.2
40.3	55	287.0	256.5	912.9	1,169.4	7.624	.13112	475.9
41.3	56	288.1	257.7	912.1	1,169.8	7.496	.13341	467.9
42.3	57	289.3	258.9	911.3	1,170.2	7.372	.13565	460.2
43.3	58	290.4	260.0	910.5	1,170.5	7.252	.13789	452.7
44.3	59	291.5	261.1	909.7	1,170.8	7.136	.14013	445.5
45.3	60	292.6	262.2	908.9	1,171.1	7.024	.14236	438.5
46.3	61	293.7	263.3	908.2	1,171.5	6.916	.14459	431.7
47.3	62	294.7	264.4	907.4	1,171.8	6.811	.14682	425.2
48.3	63	295.8	265.5	906.6	1,172.1	6.709	.14905	418.8
49.3	64	296.8	266.6	905.9	1,172.5	6.610	.15128	412.6
50.3	65	297.8	267.6	905.2	1,172.8	6.515	.15350	406.6
51.3	66	298.8	268.6	904.4	1,173.0	6.422	.15572	400.8
52.3	67	299.8	269.7	903.7	1,173.4	6.332	.15794	395.2
53.3	68	300.8	270.8	903.0	1,173.7	6.244	.16016	389.8
54.3	69	301.8	271.7	902.3	1,174.0	6.159	.16237	384.5
55.3	70	302.8	272.7	901.6	1,174.3	6.076	.16458	379.3
56.3	71	303.7	273.6	901.0	1,174.6	5.995	.16679	374.3
57.3	72	304.7	274.6	900.2	1,174.8	5.917	.16900	369.4
58.3	73	305.6	275.6	899.6	1,175.2	5.841	.17121	364.6
59.3	74	306.5	276.5	898.9	1,175.4	5.767	.17342	360.0
60.3	75	307.4	277.4	898.3	1,175.7	5.694	.17562	355.5
61.3	76	308.3	278.4	897.7	1,176.1	5.624	.17783	351.1
62.3	77	309.2	279.3	897.0	1,176.3	5.555	.18003	346.8
63.3	78	310.1	280.2	896.4	1,176.6	5.488	.18223	342.6
64.3	79	311.0	281.1	895.7	1,176.8	5.422	.18443	338.5
65.3	80	311.9	282.0	895.1	1,177.1	5.358	.18663	334.5
66.3	81	312.7	282.8	894.4	1,177.3	5.296	.18882	330.6
67.3	82	313.6	283.7	893.9	1,177.6	5.235	.19102	326.8
68.3	83	314.4	284.6	893.3	1,177.9	5.176	.19321	323.1
69.3	84	315.3	285.4	892.7	1,178.1	5.118	.19540	319.5
70.3	85	316.1	286.3	892.1	1,178.4	5.061	.19759	315.9
71.3	86	316.9	287.1	891.5	1,178.6	5.006	.19978	312.5
72.3	87	317.7	287.9	891.0	1,178.9	4.951	.20197	309.1
73.3	88	318.5	288.8	890.3	1,179.1	4.898	.20416	305.8
74.3	89	319.3	289.6	889.8	1,179.4	4.846	.20634	302.5
75.3	90	320.1	290.4	889.2	1,179.6	4.796	.20853	299.4

TABLE XX.—PROPERTIES OF SATURATED STEAM—PRESSURES, TEMPERATURE, VOLUME, WEIGHT, ETC.—(*Continued.*)

Gauge pressure, pounds.	Absolute pressure, pounds.	Temperature of water, Fahrenheit.	Heat-units from 32° F. to temp., Column 3.	Latent heat of vaporization, units.	Total heat, Columns 4 and 5, units.	Volume of 1 pound, cubic feet.	Weight of 1 cubic foot of steam.	Relative volume to water.
1	2	3	4	5	6	7	8	9
76.3	91	320.9°	291.2	888.7	1,179.9	4.746	.21071	296.3
77.3	92	321.7	292.0	888.1	1,180.1	4.697	.21289	293.2
78.3	93	322.4	292.8	887.6	1,180.4	4.650	.21507	290.2
79.3	94	323.2	293.5	887.0	1,180.5	4.603	.21725	287.3
80.3	95	323.9	294.3	886.4	1,180.7	4.557	.21943	284.5
81.3	96	324.7	295.1	885.9	1,181.0	4.513	.22160	281.7
82.3	97	325.4	295.8	885.3	1,181.2	4.469	.22378	279.0
83.3	98	326.2	296.6	884.8	1,181.4	4.426	.22595	276.3
84.3	99	326.9	297.4	884.3	1,181.7	4.384	.22812	273.7
85.3	100	327.6	298.1	883.8	1,181.9	4.342	.23029	271.1
86.3	101	328.3	298.8	883.2	1,182.0	4.302	.23246	268.5
87.3	102	329.1	299.6	882.8	1,182.3	4.262	.23463	266.0
88.3	103	329.8	300.3	882.2	1,182.5	4.223	.23680	263.6
89.3	104	330.5	301.0	881.7	1,182.7	4.185	.23897	261.2
90.3	105	331.2	301.7	881.2	1,182.9	4.147	.24114	258.9
91.3	106	331.9	302.4	880.7	1,183.1	4.110	.24330	256.6
92.3	107	332.6	303.2	880.3	1,183.5	4.074	.24547	254.8
93.3	108	333.2	303.9	879.7	1,183.6	4.038	.24763	252.1
94.3	109	333.9	304.6	879.2	1,183.8	4.003	.24979	249.9
95.3	110	334.6	305.2	878.8	1,184.0	3.969	.25195	247.8
96.3	111	335.3	305.9	878.3	1,184.2	3.935	.25411	245.7
97.3	112	335.9	306.6	877.7	1,184.3	3.902	.25626	243.6
98.3	113	336.6	307.3	877.3	1,184.6	3.870	.25842	241.6
99.3	114	337.2	308.0	876.8	1,184.8	3.838	.26058	239.6
100.3	115	337.9	308.6	876.4	1,185.0	3.806	.26273	237.6
101.3	116	338.5	309.3	875.9	1,185.2	3.775	.26489	235.7
102.3	117	339.2	309.9	875.4	1,185.3	3.745	.26704	233.8
103.3	118	339.8	310.6	875.0	1,185.6	3.715	.26920	231.9
104.3	119	340.4	311.2	874.5	1,185.7	3.685	.27135	230.1
105.3	120	341.1	311.9	874.0	1,185.9	3.656	.27350	228.3
106.3	121	341.7	312.5	873.7	1,186.2	3.628	.27565	226.5
107.3	122	342.3	313.2	873.2	1,186.4	3.600	.27780	224.7
108.3	123	342.9	313.8	872.7	1,186.5	3.572	.27995	223.0
109.3	124	343.5	314.4	872.3	1,186.7	3.545	.28210	221.3
110.3	125	344.1	315.1	871.8	1,186.9	3.518	.28424	219.6
111.3	126	344.7	315.7	871.4	1,187.1	3.492	.28639	218.0
112.3	127	345.3	316.3	871.0	1,187.3	3.466	.28853	216.4
113.3	128	345.9	316.9	870.5	1,187.4	3.440	.29068	214.8
114.3	129	346.5	317.5	870.0	1,187.6	3.415	.29282	213.2
115.3	130	347.1	318.1	869.5	1,187.8	3.390	.29496	211.6
116.3	131	347.6	318.7	868.9	1,188.0	3.370	.29700	210.1
117.3	132	348.2	319.3	868.3	1,188.2	3.355	.29900	208.6
118.3	133	348.8	319.9	867.9	1,188.3	3.340	.30060	207.1
119.3	134	349.4	320.6	867.5	1,188.5	3.328	.30220	205.7
120.3	135	350.0	321.3	867.0	1,188.7	3.304	.30580	204.2

TABLE XX.—PROPERTIES OF SATURATED STEAM—PRESSURES, TEMPERATURE, VOLUME, WEIGHT, ETC.—(*Continued.*)

Gauge pressure, pounds.	Absolute pressure, pounds.	Temperature of water, Fahrenheit.	Heat-units from 32° F. to temp., Column 3.	Latent heat of vaporization, units.	Total heat, Columns 4 and 5, units.	Volume of 1 pound, cubic feet.	Weight of 1 cubic foot of steam.	Relative volume to water.
1	2	3	4	5	6	7	8	9
121.3	136	350.5°	321.9	867.6	1,188.9	3.280	.30840	202.8
122.3	137	351.1	322.5	867.1	1,189.0	3.260	.31045	201.4
123.3	138	351.8	323.1	866.6	1,189.2	3.240	.31292	200.0
124.3	139	352.2	323.7	866.1	1,189.4	3.220	.31313	198.7
125.3	140	352.8	324.3	865.6	1,189.6	3.201	.31534	197.3
126.3	141	353.3	325.0	865.1	1,189.7	3.182	.31752	196.0
127.3	142	353.9	325.7	864.5	1,189.9	3.163	.31950	194.7
128.3	143	354.4	326.3	863.9	1,190.0	3.144	.32110	193.4
129.3	144	355.0	327.1	863.2	1,190.2	3.123	.32320	192.2
130.3	145	355.5	327.8	862.6	1,190.4	3.101	.32530	190.9
131.3	146	356.0	328.4	862.2	1,190.5	3.067	.3274	189.7
132.3	147	356.6	328.9	861.8	1,190.7	3.050	.3295	188.5
133.3	148	357.1	329.5	861.4	1,190.9	3.03	.3316	187.3
134.3	149	357.6	330.0	861.0	1,191.0	3.01	.3337	186.1
135.3	150	358.2	330.6	860.6	1,191.2	2.99	.3358	184.9
136.3	151	358.7	331.1	860.2	1,191.3	2.97	.3379	183.7
137.3	152	359.2	331.6	859.9	1,191.5	2.95	.3400	182.6
138.3	153	359.7	332.2	859.5	1,191.7	2.93	.3421	181.5
139.3	154	360.2	332.7	859.1	1,191.8	2.91	.3442	180.4
140.3	155	360.7	333.2	858.7	1,192.0	2.89	.3463	179.2
141.3	156	361.1	333.8	858.4	1,192.1	2.87	.3483	178.1
142.3	157	361.8	334.3	858.0	1,192.3	2.85	.3504	177.0
143.3	158	362.3	334.8	857.6	1,192.4	2.84	.3525	176.0
144.3	159	362.8	335.3	857.2	1,192.6	2.82	.3546	174.9
145.3	160	363.3	335.9	856.9	1,192.7	2.80	.3567	173.9
146.3	161	363.8	336.4	856.5	1,192.9	2.79	.3588	172.9
147.3	162	364.3	336.9	856.1	1,193.0	2.77	.3609	171.9
148.3	163	364.8	337.4	855.8	1,193.2	2.76	.3630	171.0
149.3	164	365.3	337.9	855.4	1,193.3	2.74	.3650	170.0
150.3	165	365.7	338.4	855.1	1,193.5	2.72	.3671	169.0
151.3	166	366.2	338.9	854.7	1,193.6	2.71	.3692	168.1
152.3	167	366.7	339.4	854.4	1,193.8	2.69	.3713	167.1
153.3	168	367.2	339.9	854.0	1,193.9	2.68	.3734	166.2
154.3	169	367.7	340.4	853.6	1,194.1	2.66	.3754	165.3
155.3	170	368.2	340.9	853.3	1,194.2	2.65	.3775	164.3
156.3	171	368.6	341.4	852.9	1,194.4	2.63	.3796	163.4
157.3	172	369.1	341.9	852.6	1,194.5	2.62	.3817	162.5
158.3	173	369.6	342.4	852.3	1,194.7	2.61	.3838	161.6
159.3	174	370.0	342.9	851.9	1,194.8	2.59	.3858	160.7
160.3	175	370.5	343.4	851.6	1,194.9	2.58	.3879	159.8
161.3	176	371.0	343.9	851.2	1,195.1	2.56	.3900	158.9
162.3	177	371.4	344.3	850.9	1,195.2	2.55	.3921	158.1
163.3	178	371.9	344.8	850.5	1,195.4	2.54	.3942	157.2
164.3	179	372.4	345.3	850.2	1,195.5	2.52	.3962	156.4
165.3	180	372.8	345.8	849.9	1,195.7	2.51	.3983	155.6

TABLE XX.—PROPERTIES OF SATURATED STEAM—PRESSURES, TEMPERATURE, VOLUME, WEIGHT, ETC.—(*Continued.*)

Gauge pressure, pounds.	Absolute pressure, pounds.	Temperature of water, Fahrenheit.	Heat-units from 32° F. to temp.; Column 3.	Latent heat of vaporization, units.	Total heat, Columns 4 and 5, units.	Volume of 1 pound, cubic feet.	Weight of 1 cubic foot of steam.	Relative volume to water.
1	2	3	4	5	6	7	8	9
166.3	181	373.3°	346.3	849.5	1,195.8	2.50	.4004	154.8
167.3	182	373.7	346.7	849.2	1,195.9	2.48	.4025	154.0
168.3	183	374.2	347.2	848.9	1,196.1	2.47	.4046	153.2
169.3	184	374.6	347.7	848.5	1,196.2	2.46	.4066	152.4
170.3	185	375.1	348.1	848.2	1,196.3	2.45	.4087	151.6
171.3	186	375.5	348.6	847.9	1,196.5	2.43	.4108	150.8
172.3	187	375.9	349.1	847.6	1,196.6	2.42	.4129	150.0
173.3	188	376.4	349.5	847.2	1,196.7	2.41	.4150	149.2
174.3	189	376.9	350.0	846.9	1,196.9	2.40	.4170	148.5
175.3	190	377.3	350.4	846.6	1,197.0	2.39	.4191	147.8
176.3	191	377.7	350.9	846.3	1,197.1	2.37	.4212	147.0
177.3	192	378.2	351.3	845.9	1,197.3	2.36	.4233	146.3
178.3	193	378.6	351.8	845.6	1,197.4	2.35	.4254	145.6
179.3	194	379.0	352.2	845.3	1,197.5	2.34	.4275	144.9
180.3	195	379.5	352.7	845.0	1,197.7	2.33	.4296	144.2
181.3	196	380.0	353.1	844.7	1,197.8	2.32	.4317	143.5
182.3	197	380.3	353.6	844.4	1,197.9	2.31	.4337	142.8
183.3	198	380.7	354.0	844.1	1,198.1	2.29	.4358	142.1
184.3	199	381.2	354.4	843.7	1,198.2	2.28	.4379	141.4
185.3	200	381.6	354.9	843.4	1,198.3	2.27	.4400	140.8
190.3	205	383.7	357.1	841.9	1,199.0	2.22	.4503	137.5
195.3	210	385.7	359.2	840.4	1,199.6	2.17	.4605	134.5
200.3	215	387.7	361.3	838.9	1,200.2	2.12	.4707	131.5
205.3	220	389.7	362.2	838.6	1,200.8	2.06	.4852	128.7
215.3	230	393.6	366.2	835.8	1,202.0	1.98	.5061	123.3
225.3	240	397.3	370.0	833.1	1,203.1	1.90	.5270	118.5
235.3	250	400.9	373.8	830.5	1,204.2	1.83	.5478	114.0
285.3	300	417.4	390.9	818.3	1,209.2	1.535	.6515	95.8
335.3	350	432.0	406.3	807.5	1,213.7	1.325	.7545	82.7
385.3	400	444.9	419.8	797.9	1,217.7	1.167	.8572	72.8
435.3	450	456.6	432.2	789.1	1,221.3	1.042	.9595	65.1
485.3	500	467.4	443.5	781.0	1,224.5	.942	1.062	58.8
535.3	550	477.5	454.1	773.5	1,227.6	.859	1.164	53.6
585.3	600	486.9	464.2	766.3	1,230.5	.790	1.266	49.3
635.3	650	495.7	473.6	759.6	1,233.2	.731	1.368	45.6
685.3	700	504.1	482.4	753.3	1,235.7	.680	1.470	42.4
735.3	750	512.1	490.9	747.2	1,238.0	.636	1.572	39.6
785.3	800	519.6	498.9	741.4	1,240.3	.597	1.674	37.1
835.3	850	526.8	506.7	735.8	1,242.5	.563	1.776	34.9
885.3	900	533.7	514.0	730.6	1,244.7	.532	1.878	33.0
935.3	950	540.3	521.3	725.4	1,246.7	.505	1.980	31.4
985.3	1,000	546.8	528.3	720.3	1,248.7	.480	2.082	30.0

CHAPTER X

FLOW OF STEAM THROUGH ORIFICES, NOZLES, AND PIPES

THE flow of steam from an orifice into a vacuum may be computed with approximate accuracy by the formula:

(12) . . . $\sqrt{T+460.6}\times 60.2$, or the square root of the absolute temperature multiplied by 60.2, and the product by .54, the coefficient for the velocity at initial density for an orifice.

For example: at 75 pounds absolute pressure, 60 pounds gauge-pressure, the temperature is 307.4, and

$$\sqrt{307.4+460.6}=\sqrt{768}=27.712\times 60.2=1{,}667\times .54=899.7,$$

velocity of steam at its initial density.

Into the atmosphere, steam flows through a thin plate-orifice, at its initial density due to its absolute pressure, with a velocity of 3.5953 $\sqrt{\text{height}}$ in feet of a column of steam, uniform in density, equal to its weight at its initial pressure per square foot. The height is equal to the volume of 1 pound of steam, as in Column 7, Table XX, of the properties of saturated steam, multiplied by 144 square inches in 1 square foot. For example:

For 100 pounds absolute pressure it is $100\times 4.342\times 144=62{,}524.8$ feet, and $\sqrt{62{,}524.8}=250.7\times 3.5953=901.3$ feet per second.

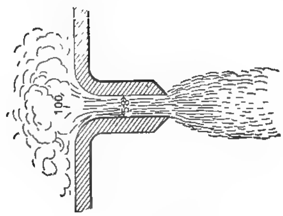


FIG. 120.—Straight nozle.

The velocity of the jet of steam from an orifice or nozle is increased in the ratio of 1.624, so that in a short straight nozle, Fig. 120, of from two to two and one-half times its diameter in length, with good entrance-curves, the velocity may be $901\times 1.624=1{,}463$ feet. In the expanding nozles, as designed for steam-turbines of the Delaval class, a much higher velocity is claimed.

The velocity of flow of steam in pipes depends upon the pressure-head, which is the height in feet of a column of steam of a uniform

density of the steam at the entrance of the pipe—the length and diameter of the pipe in feet—a fractional exponent—and the head against which it is flowing at the terminal.

The formula much in use for steam flowing from a long pipe into the atmosphere is:

(13) . . . $50\sqrt{\frac{h}{L}\,d}$ = velocity in feet per second.

L and d = length and diameter of the pipe in feet or decimals of a foot for d.

Then, as in the previous example, for 100 feet in length of a 1-inch pipe, and boiler-pressure of 100 pounds absolute, 85.3 gauge, the height h, as before explained, is 62,524.8, and

$$\sqrt{\frac{62,524.8}{100}\times .0833}=\sqrt{70.6}=8.402\times 50=400.1 \text{ feet per second.}$$

The acceleration in velocity of steam by its expansive effort in passing from a converging section to and through a diverging section of a jet-nozle, such as used for impulse energy in steam-turbines,

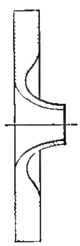


FIG. 121.—Steam-nozle.

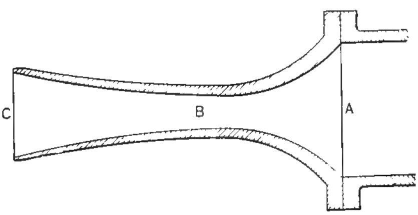


FIG. 122.—Expanding nozle.

is due to its expansive volume being greater than the increasing area of the diverging walls of the nozle, less the loss by cooling, as the expansion is adiabatic. From the general formula $\sqrt{T+460.6}\times 60.2$ for 75 pounds absolute pressure, the velocity at initial pressure gives 1,667 feet per second in the throat of an expanding nozle. Then the relative volumes per pound, of steam at 75 pounds and atmospheric pressure, is: $\frac{26.37}{5.69}=4.63$; and if the relative areas of the expanding nozle are as 1 to 2, the volume at the mouth of the nozle

will be: $\frac{4.63}{2}=2.315$, and deducting the shrinkage from the adiabatic expansion, $2.315^{1.3}=1.78$; then $1,667\times1.78=2,968$ feet per second.

Fig. 123 shows the diverging forms of nozles for impact-wheels, in which the angle of impact should be as near 20° from the plane of rotation as possible.

It is claimed, as stated by D. K. Clark, that when steam flows from a nozle of the best form, "the velocity does not increase when flowing into a resisting medium at any pressure below 58 per cent.

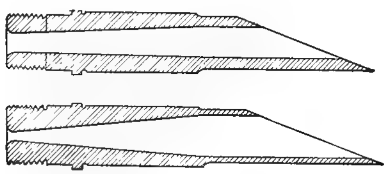


FIG. 123.—Diverging nozles.

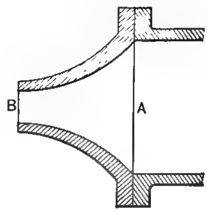


FIG. 124.—Nozle of best form.

of the initial pressure." Then 160 pounds absolute pressure $\times58$ per cent. $=92.8$ pounds absolute, the lowest resisting pressure at which the velocity ceases to increase.

The ratio of the volumes at these pressures is: $\frac{4.6}{2.8}=1.643$. Then, using the temperature formula $\sqrt{T+460.6}\times60.2$ for 160 pounds absolute pressure, we have

$$28.7\times 60.2=1,727.7\times.58=1,002\times1.643=1,646 \text{ feet per second.}$$

Again using the formula $3.5953\ \sqrt{\text{height}}$, we have $160\times2.8\times144=64,512$ feet, and

$\sqrt{64,512}=253.9\times3.5953=912.8$, and $912.8\times1.643=1,499$ feet per second.

In a diverging continuation of the nozle, as used for steam-turbines of the Delaval type, the acceleration in velocity due to expansion, less the adiabatic condition of expansion, will be in the ratio of the expanding-nozle areas at initial and terminal ends. Then if the ratio is 1 to 2, the velocity from the above equation should be: $1,499\times2=2,998$ feet per second.

A formula deduced from Professor Rateau's formula, based on the area of the entropy diagram for the velocity of steam in feet per second, is:

$$(14) \quad . \quad . \quad V=224\sqrt{(T_1-T_2)\left(\frac{r}{T_1}+\frac{T_1-T_2}{T_1+T_2}\right)}, \text{ in which}$$

$$224=\sqrt{2g\times 778}.$$

T_1 = absolute initial temperature of saturated steam; T_2 = absolute terminal temperature at, say, 100 + 460.6° F.; r = latent heat of vaporization at temperature T_1, as in Column 5 of steam table. Then from a nozle of best form with steam expanding from an initial pressure of 160 pounds absolute into a vacuum of a little less than 1 pound absolute pressure, or 28 inches of mercury, at which pressure the absolute temperature $T_2 = 100 + 460.6 = 560.6$, and substituting figures for the letters in the formula, we have for 160 pounds absolute:

$$224\sqrt{823.9-560.6\left(\frac{856.9}{824.2}+\frac{263.6}{1{,}384.8}\right)}=4{,}027, \text{ the velocity in feet per}$$

second.

The following table has been computed for velocities from a pressure of 160 pounds absolute, expanded to various stages of lower pressure by Rateau's formulas, in which the figures in the tables of properties of saturated steam were used. For the dryness of steam from condensation, 1 = dry saturated steam, and $1-x$ = the percentage of moisture or condensation by expansion.

This is found by the formula:

$$(15) \quad . \quad . \quad . \quad \frac{T_2}{\text{l. h. } T_2}\left(\frac{\text{l. h. } T_2}{T_1}+\text{hy. log. }\frac{T_1}{T_2}\right)=x, \text{ the relative amount}$$

of dry steam after expansion, in which T_2 is absolute temperature after expansion; T_1 absolute initial temperature; l. h. $T_{1\cdot 2}$, latent heat of vaporization, as in Column 5, steam table XX.

For example, substituting the values for expansion from 160 pounds absolute to atmospheric pressure, we have:

$$\frac{672.6}{966.1}\left(\frac{856.9}{824.2}+\text{hy. log. }\frac{824.2}{672.6}\right)=x, \text{ as in Table XXI.}$$

For the area of a nozle of best form, as in Column 9 of Table XXI, the formula is: $\frac{x\ 144}{DV}$ (Column 8), multiplied by the square inches in a foot, and the product divided by the product of the density and

TABLE XXI.—THEORETICAL VELOCITY AND AREAS OF AN EXPANDING NOZLE OF BEST FORM FOR DRY STEAM, EXPANDED FROM 160 POUNDS ABSOLUTE PRESSURE PER SQUARE INCH TO LOWER PRESSURES.

Pressure absol'te, lbs. per sq. in.	T_2, absolute degrees F.	T_1-T_2, degrees F.	D, lbs. per cu. ft.	Vel., ft. per sec.	D V	$\frac{p}{P}$	x, per cent. dry.	A, area of nozle, sq. in.	Profile of nozle.
1	2	3	4	5	6	7	8	9	10
160	823.9	....	.3567	....	....	......	1.000	1.000	2.31
150	818.8	5.1	.3358	519	174.2	.937	.992	.818	1.86
140	813.4	10.5	.3153	745	234.8	.875	.987	.609	1.38
130	807.7	16.2	.2949	916	270.1	.812	.982	.524	1.21
120	801.7	22.2	.2735	1,088	297.5	.750	.975	.472	1.09
110	795.2	28.7	.2519	1,236	311.3	.687	.969	.448	1.03
100	788.2	35.7	.2303	1,383	318.5	.625	.963	.444	1.02
90	780.7	43.2	.2085	1,525	317.9	.562	.957	.433	1.00
70	763.4	60.5	.1645	1,813	298.3	.437	.942	.454	1.04
50	741.5	82.4	.1199	2,129	255.1	.312	.923	.521	1.20
30	710.9	113.0	.0742	2,525	187.3	.187	.894	.740	1.70
15	673.7	150.2	.0386	2,934	112.9	.094	.865	1.103	2.54
5	623.0	200.9	.0137	3.412	47.3	.031	.817	2.489	5.72
2	586.9	237.0	.0058	3,752	21.7	.012	.786	5.202	12.01
1	562.6	261.3	.00303	4,027	12.2	.006	.769	9.07	20.94

velocity in Column 6. The weight of steam discharged per second per square inch of area will be: $\frac{DV}{x\ 144}$ in pounds.

The proportional diameters of an expanding nozle for the theoretical expansion of steam, from 160 pounds absolute, is given in Column 10, and its form shown in Fig. 125, which is a profile of an expanding

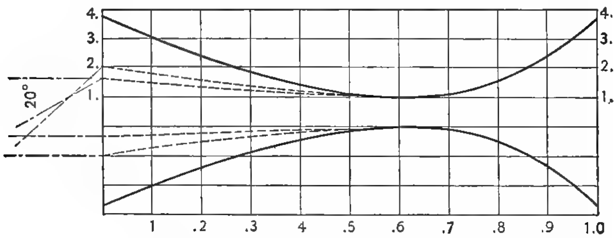


FIG. 125.—Theoretical curves of expanding nozles.

nozle to meet the conditions of velocity and expansion. In practice, the expanding part of the nozle is much reduced in order to increase its velocity by reducing the lateral expansion. Fig. 125 shows the theoretical profile of an expanding nozle in which the practical lines of the expanding part are shown by the dotted lines as used for turbine-nozles.

ENERGY OF STEAM

The theoretical energy of steam in foot-pound power is due to the difference in the heat-unit values between which it is expanded, and which, multiplied by 778, the foot-pound value per heat-unit, equals the total foot-pound value due to expansion per pound of steam. For example: the heat-units in 1 pound of steam at 160 pounds absolute are 1,192.7, and the heat-units in 1 pound at atmospheric pressure are 966.1; then, $1{,}192.7-966.1=226.6\times 778=176{,}294$ foot-pounds, and $\frac{176{,}294}{33{,}000}=5.34$ horse-power per pound of steam per minute.

FLOW OF STEAM THROUGH LONG PIPES

The weight and volume of steam which will flow through a pipe in one minute from a given pressure, and any designated loss of pressure from friction, may be obtained from the following general formula for the flow of gases and vapors:

$$\text{(16)} \quad . \quad . \quad . \quad . \quad . \qquad W=87\sqrt{\frac{D(p_1-p_2)d^5}{L\left(1+\frac{3.6}{d}\right)}}$$

W = total weight in pounds, which, divided by the weight of 1 cubic foot = cubic feet per minute; D = density or weight per cubic foot at initial pressure, p_1; p_2 = terminal pressure at end of pipe; d = actual diameter of the pipe in inches; L = length of pipe in feet.

The following table represents the weight of steam that will flow per minute through a straight, smooth pipe of 240 times its internal diameter, with a loss of 1 pound in the pressure:

For sizes of pipe less than 6 inches, the flow is calculated from the *actual* areas of "standard" pipe of such nominal diameters.

For horse-power, multiply the figures in the table by 2. For any other loss of pressure, multiply by the square root of the given loss. For any other length of pipe, *divide* 240 *by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient,* which will give the flow for 1 pound loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

TABLE XXII.—FLOW OF STEAM THROUGH PIPES OF 240 TIMES THEIR DIAMETER, WITH A LOSS OF 1 POUND IN PRESSURE.

Initial pressure by gauge. Pounds per square inch.	Diameter of pipe in inches.					Length of each = 240 diameters.					
	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12
	Weight of steam per minute in pounds, with 1 pound loss of pressure.										
1	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4
10	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5
20	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3
30	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5	824.1
40	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0
50	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5
60	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1,046.7
70	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1,108.5
80	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1,166.1
90	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1,219.8
100	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1,270.1
120	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1,363.3
150	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1,009.2	1,486.5

The loss of head due to the friction of the steam entering the pipe, and passing elbows and valves, will reduce the flow given in the table. The resistance at the opening, and that at a globe-valve, are each about the same as that for a length of pipe equal to 114 diameters divided by a number represented by $1+(3.6\div\text{diameter})$. For the sizes of pipes given in the table, these corresponding lengths are:

1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12
25	34	41	47	52	60	66	71	79	84	88

The resistance at an elbow is equal to two-thirds that of a globe-valve. These equivalents—for opening, for elbows, and for valves—must be added in each instance to the actual length of pipe. Thus a 4-inch pipe, 120 diameters (40 feet) long, with a globe-valve and three elbows, would be equivalent to $120+60+60+(3\times40)=360$ diameters long; and $360\div240=1\frac{1}{2}$. It would therefore have $1\frac{1}{2}$ pounds loss of pressure at the flow given in the table, or deliver $(1\div\sqrt{1\frac{1}{2}}=.816)=81.6$ per cent. of the steam with the same (1 pound) loss of pressure.

CHAPTER XI

SUPERHEATED STEAM AND ITS WORK

SATURATED steam, or steam which has exactly the temperature due to its pressure, has aptly been described as steam saturated with heat, and the chief peculiarity which it possesses is that the slightest abstraction of heat is followed by a corresponding condensation.

Superheated steam is generated by the addition of heat to saturated steam. The behavior of superheated steam is similar to that of gases; it is a poor conductor of heat, and has the special peculiarity of losing a certain amount of heat without becoming saturated or wet steam. The specific heat of steam is only 0.48, and therefore very little heat is required to superheat steam; but as the steam loses the heat as quickly as it acquires it, every passage conveying superheated steam should be well covered with non-conducting material. Although there are some losses on account of the heat-radiation when using superheated steam, they are very much smaller per volume, because the loss of heat from superheated steam has lower calorific value than the latent heat of saturated steam.

The economy effected by using superheated steam in engines is remarkable, and, acknowledging this fact, a great number of steam-users superheat the steam, although in many cases only a few degrees; yet a considerable saving in steam and coal is always the result. To obtain the full benefit, the required temperature of superheat should be 600° F., and to stand this temperature the engines should be specially designed.

The use of highly superheated steam does not require high boiler-pressures; 160 pounds is the highest to be recommended, as no advantage can be derived by exceeding this. As the amount of heat transmitted from the steam to cylinder-walls, and *vice versa*, is much lower with superheated steam than with saturated steam, the whole range of temperature from boiler-pressure to vacuum can take place in two cylinders, so that the use of a triple-expansion engine does not make very much improvement in economy.

In view of the great advantages of steam-superheating, and the great number of engines running at present satisfactorily, it is astonishing that a few failures have caused prejudices among some engineers, who make the general introduction of the use of superheated steam very difficult. It will be worth mentioning that the results of a great number of trials have always proved a great saving in steam and coal, and even with small plants and simple piston-valve engines almost the same good economy is obtainable as with large engines with most exact valve-gears. It is therefore recommended that superheated steam should be used in connection with all engines; the only question to be settled is the degree of superheat, which largely depends on local circumstances and the construction of the engine.

Superheated-steam engines use on an average 30 to 40 per cent. less steam than saturated-steam engines of the same type. Consequently boilers can be made 30 per cent. smaller, and the difference in price will nearly cover the cost of the superheater. For the same steam-consumption the superheated-steam engine is cheaper, as it may be worked with a lower boiler-pressure.

One of the most troublesome effects of expansion is found in the action of the steam-valves. Slide-valves and Corliss valves are naturally affected by the high temperatures, 480 to 500° F. being the upper limit for the latter. Piston-valves, when carefully constructed and proportioned, answer well, but they must be made especially for the service. The longitudinal expansion of the cylinder tends to deform the steam-chest and valve-seat, and provision must be made for such effects. Rings and springs in valves are objectionable, as it is difficult to keep the steam from getting between the rings and creating increased pressure and friction. Poppet-valves have been used with much success.

The adoption of superheated steam in steam-engines was made possible by the manufacture of heavy mineral oils with high-ignition points and by the now common practice of using metallic packing. From a purely theoretical point of view the advantage gained is small, and if the conditions were those of the ideal engine, superheating would never have been heard of. On entering the cylinder of a steam-engine part of the steam is condensed, without doing any work, by coming in contact with the walls and piston, cooled from the previous exhaust-

stroke. This reduces the working value of the steam and coats the cylinder with a film of water, which conducts the heat much more readily to the cylinder-walls than in the case of dry steam.

The principal object of superheating is to reduce this transfer of heat and initial condensation; for although the superheated steam gives up some of its heat to the metal on admission, there is no condensation, the only effect being a reduction of volume and a fall in temperature. Superheating also tends to prevent leakage at sliding surfaces, such as piston-rings, valves, etc. No matter how tight they are when at rest, a film of water, creeping along between the sliding surfaces, will cause steam to leak through when the engine is running.

At 300° F. of superheat the volume of steam is increased about 50 per cent., and owing to this increase in volume less heat is required to produce the same volume for superheated steam than for saturated steam at the same pressure. Thus less heat enters the cylinders at each stroke, and as the same amount of heat is converted into work in each case, the economy of superheating is apparent.

It has been found that to attain a certain velocity of steam in a pipe, superheated steam requires a smaller drop in pressure than saturated steam and, as less steam is required per horse-power when superheated, a reduction may be made in the size of the piping, which again will reduce the cost as well as the loss from radiation.

A separator will be unnecessary, which also reduces the radiating surface; and the absence of water in the steam-pipes does away with all risk of getting water into the cylinders.

The superheater may have three different positions relative to the boiler: 1. It may be placed in the flue so as to extract heat from the gases as they leave the boiler, which is the most economical method of obtaining superheat. 2. It may be placed in the path of the gases between the fire and the boiler proper. 3. It may be quite separate, and independently fired.

In case 1, a good boiler should take up enough of the heat from the gases to allow them to pass out at a temperature but little above that of the boiler. If only a low degree of superheat is required, this position is much the simplest and cheapest.

Case 2 is the most economical method from point of view of fuel required, but the difficulty of regulating the temperature is much

greater. In Case 3 the temperature can be easily regulated and the superheater readily cut out when required. It is a more wasteful method than either of the other two, but as little coal is required by the superheater when compared with the boiler, the loss does not count for much.

It may be taken as a general rule that the better the economy of an engine, the less gain there will be from superheat. Thus the best results should be looked for in a simple engine with a low steam-pressure.

The question of the cost of the additional heat is sometimes raised, but it can be shown that much less heat is required to produce a cubic foot of superheated steam than to generate a cubic foot of saturated steam of the same pressure.

In triple and quadruple expansion the gain is small unless the superheating is carried to a high temperature. It varies, of course, with the point of cut-off and the ratio of expansion.

There is little gain due to superheating in any cylinder after the stage is reached in which the steam is kept dry throughout the whole expansion, the only object of taking the superheat higher is to obtain dry steam as far as possible during the whole expansion of the engine.

The same effect may be obtained by reheating the steam in the receivers between the different cylinders, so that it will enter each cylinder superheated sufficiently to insure dryness at the end of the stroke in that cylinder. This can be effected by passing the superheated steam through coils in the receiver, so as to give up part of its heat to the steam that has already expanded in the previous cylinder and pass on to the steam-chest of the engine at a lower degree of superheat. This method gives the advantage of a high degree of superheat without the disadvantage of extremely high temperature in the cylinders.

The effect of superheating in turbines is somewhat different from that in engines. In some turbines the potential energy of the steam is transformed into kinetic energy before being available for doing work. This is effected by means of a large drop of pressure through a small nozle. Some of the heat appears as kinetic energy, and if saturated steam were used, there would be water present at this pressure. If superheated, the extra heat in the steam would prevent any condensation and increase the volume and velocity of the steam.

With this form of turbine the high-temperature steam never comes in contact with the frictional parts, and the only limit to the degree of superheat is apparently the temperature at which the strength of steel begins to be affected. Results of a trial have been published on a De Laval turbine using steam at 930° F., with no serious difficulties encountered.

The ideal efficiency of a heat-engine is determined by the range of temperature through which it works and not by the medium through which that heat is used. This theoretical fact is employed as an argument in favor of superheated steam. It is pretty generally recognized now that one of the losses of the steam-engine is the interchange of heat which goes on between the cylinder metal and the working steam. To prevent this interchange it is usual to superheat the steam so that less of it may turn into water, for it is in the form of water that the working fluid exerts its worst effects.

Theorists who look on superheat as a means of raising the temperature of the working fluid, overlook some important practical considerations. During the period of time that the admission-port of the cylinder is open, the piston of the engine is pushed forward by the pressure of the steam in the boiler. The steam in the pipe does not expand. It flows into the cylinder in obedience to the push which it receives from behind, and this push is not even due in all cases or entirely to expansion in the boiler. It is due directly to the heat of the fire, which causes the water to turn into steam. It is this bulk of new steam that pushes the engine-piston, and the steam between the boiler and the moving face of the piston is simply a strut; for, since it maintains a constant pressure, it cannot expand. In its passage from the boiler to the engine through a superheater it receives additional heat per given volume of saturated steam, and expands to a new volume, due to the amount of superheat, without receiving any addition of water, and thereby assumes the condition of a gas. As a gas it does work by expansion, without loss from condensation, until its temperature falls to the saturation-point, when its further expansion assumes the condition of saturated steam.

Therefore there can be no practical economy, considering the heat-troubles from wear and tear, by using superheat at a greater temperature than will insure dry steam to the end of its expansion-work.

TABLE XXIII.—SPECIFIC VOLUME OF SUPERHEATED STEAM IN CUBIC FEET PER POUND AT TEMPERATURES ABOVE THAT OF SATURATED STEAM.

Absolute pressure.	Saturated steam, volume.	Specific volume for degrees of superheat, Fahrenheit.									
		20°	40°	60°	80°	100°	120°	140°	160°	180°	200°
70	6.14	6.47	6.64	6.81	6.98	7.15	7.32	7.49	7.66	7.83	8.00
80	5.42	5.72	5.88	6.03	6.17	6.32	6.47	6.62	6.77	6.92	7.07
90	4.86	5.15	5.28	5.41	5.54	5.67	5.81	5.94	6.07	6.20	6.33
100	4.04	4.67	4.79	4.91	5.03	5.15	5.27	5.39	5.51	5.63	5.75
110	4.03	4.29	4.42	4.51	4.61	4.72	4.83	4.94	5.05	5.15	5.26
120	3.71	3.96	4.06	4.16	4.26	4.36	4.46	4.56	4.66	4.75	4.85
130	3.44	3.69	3.78	3.87	3.96	4.05	4.14	4.23	4.32	4.41	4.51
140	3.21	3.45	3.53	3.62	3.69	3.79	3.87	3.96	4.05	4.13	4.20
150	3.01	3.24	3.32	3.40	3.48	3.55	3.63	3.71	3.79	3.87	3.95
160	2.83	3.05	3.13	3.20	3.28	3.36	3.42	3.50	3.57	3.64	3.72
170	2.67	2.89	2.96	3.03	3.10	3.17	3.24	3.31	3.38	3.45	3.52
180	2.53	2.75	2.81	2.88	2.94	3.01	3.07	3.14	3.21	3.28	3.34
190	2.41	2.62	2.68	2.74	2.80	2.87	2.93	2.99	3.05	3.12	3.18
200	2.29	2.50	2.56	2.62	2.68	2.74	2.80	2.86	2.91	2.97	3.03

The above table has been computed by Schmidt's formula based on Hirns's experiments, namely:

(17) . . . . . . $Sv = 0.59276 \times \frac{441.4 + T}{P}$,

in which Sv = specific volume in cubic feet per pound, T = temperature of saturated steam + superheat, P = absolute pressure in pounds per square inch. The percentage of increase in volume from superheat, apart from its freedom from condensation, is the most essential factor of economy from the use of superheat; for instance, the volume of saturated steam at 160 pounds is 2.83 cubic feet per pound, and is increased by 200° F. superheat to 3.72 cubic feet per pound, and $\frac{3.72}{2.83} = 1.31$, or 31 per cent. increase in volume; and at only 100° of superheat, which may be saved from the chimney-gases, the increase in volume is $\frac{3.36}{2.83} = 1.116$, or over 11 per cent.

The following table represents a fair approximation to ordinary practice, but does not meet the extraordinary tests that have been published for short runs with superheat reaching near or quite to the temperature of incandescence. Such tests may make a good showing, but are not practicable for continued service. Mineral oils will not give the required service at temperatures above their boiling-point.

TABLE XXIV.—ESTIMATED STEAM-CONSUMPTION WITH SUPERHEAT OF 100°, 200°, AND 300° F., FOR VARIOUS PRESSURES AND CUT-OFFS IN COMPOUND CONDENSING-ENGINES. 27-INCH VACUUM.

BY W. O. WEBBER.

Initial pressure.	One-Fifth Cut-Off.					One-Fourth Cut-Off.					One-Third Cut-Off.				
	M. E. P.	Pounds steam per I. H. P. hour.	Steam with superheat at			M. E. P.	Pounds steam per I. H. P. hour.	Steam with superheat at			M. E. P.	Pounds steam per I. H. P. hour.	Steam with superheat at		
			100°	200°	300°			100°	200°	300°			100°	200°	300°
200	96.2	17.05	15.00	13.20	11.60	112.3	17.86	15.73	13.85	12.20	134.8	18.67	16.41	14.45	12.70
190	91.0	17.14	15.07	13.27	11.65	106.3	17.95	15.80	13.92	12.26	127.8	18.76	16.50	14.52	12.76
180	85.8	17.25	15.16	13.34	11.72	100.3	18.05	15.87	13.98	12.31	120.8	18.85	16.58	14.59	12.82
170	80.5	17.32	15.24	13.41	11.77	94.4	18.13	15.95	14.04	12.36	113.8	18.95	16.67	14.66	12.88
160	75.3	17.42	15.32	13.49	11.84	88.4	18.23	16.02	14.11	12.42	106.3	19.04	16.75	14.73	12.95
150	70.1	17.50	15.40	13.57	11.91	82.4	18.32	16.12	14.18	12.47	99.8	19.14	16.84	14.81	13.03
140	64.9	17.60	15.48	13.65	11.97	76.5	18.37	16.20	14.25	12.53	92.8	19.24	16.92	14.88	13.10
130	59.7	17.70	15.57	13.72	12.04	70.5	18.52	16.27	14.32	12.59	85.8	19.34	17.02	14.96	13.17
120	54.4	17.80	15.67	13.80	12.11	64.5	18.63	16.37	14.40	12.65	78.8	19.44	17.12	15.04	13.24
110	49.2	17.90	15.76	13.90	12.18	58.6	18.73	16.46	14.48	12.73	71.7	19.55	17.22	15.12	13.31
100	44.0	18.02	15.85	13.98	12.26	52.6	18.85	16.56	14.56	12.80	64.7	19.66	17.32	15.21	13.38
90	38.8	18.13	15.96	14.07	12.34	46.6	18.95	16.67	14.66	12.88	57.7	19.80	17.42	15.30	13.47
80	33.6	18.26	16.07	14.17	12.43	40.7	19.08	16.80	14.76	12.97	50.7	19.90	17.52	15.39	13.55
70	28.4	18.40	16.20	14.27	12.53	34.7	19.22	16.93	14.87	13.07	43.6	20.04	17.64	15.50	13.64
60	23.1	18.55	16.33	14.37	12.64	28.7	19.37	17.07	15.00	13.18	36.6	20.18	17.77	15.51	13.74
50	17.9	18.74	16.48	14.50	12.76	22.8	19.55	17.24	15.16	13.32	29.6	20.35	17.90	15.73	13.84

The superheat of from 400° to 700° F. seems practically absurd, and the claim of less than 10 pounds of steam per horse-power is only suited to an experimental test.

The saving in steam by superheat, as shown in Table XXIV, say, for example, at 160 pounds, with the varying mean pressures due to cut-off, is, for 200° F. superheat at $\frac{1}{5}$ cut-off $=\frac{17.42}{13.49}=1.291$; at $\frac{1}{4}$ cut-off $\frac{18.23}{14.11}=1.294$; at $\frac{1}{3}$ cut-off $\frac{19.04}{14.73}=1.292$, showing a uniform increase of volume of 29 per cent. for various degrees of cut-off at 200° F. superheat. At 300° F. the increase in volume is 47 per cent.

Superheating should not be regarded as a means of carrying more heat to an engine, but only as a preventive of waste through condensation. It has been proved by experiment that about 8° F. of superheat are required to prevent each 1 per cent. of moisture in the cylinder at cut-off when using saturated steam. If the specific heat of superheated steam at constant pressure be taken as 0.48, it follows that a rise of 8° F. in the temperature above the normal temperature of saturated steam of the same pressure represents the expenditure of $0.48\times8=3.84$ thermal units. Assuming the initial condensation of the entering steam to be about 20 per cent., then $3.84\times20=76.8$ thermal units which must be added in the form of superheat to insure dry steam at cut-off.

The amount of fuel required to superheat steam, and the quantity of fuel that must be burned to continue this heat, are greater than is commonly supposed. It takes, as will be noticed in steam table XX, approximately 1,100 thermal units to convert a pound of feed-water at ordinary temperatures into steam at the usual temperature. By the addition of 76.8 thermal units in the form of superheat, we have increased the expenditure of heat by about 7 per cent. If all the 20 per cent. of condensation is saved there is undoubtedly a decided gain; and this fact is true, that a small amount of superheat is desirable in all forms of engines. It is in the higher degrees of superheat that this difference vanishes, because the specific heat of superheated steam increases with the degree of superheat.

The specific heat at constant pressure, C_p, of superheated steam at atmospheric pressure and near the point of saturation was found by

Regnault to be 0.48, and until recently this value was thought to apply to the specific heat at higher pressures. It probably varies, as does the specific heat at constant volume, C_v, which has been assigned a slightly decreasing value for superheat with increasing pressure, as follows:

Pressure	50	100	200	300
C_v	0.348	0.346	0.344	0.341

for steam of moderate superheat. By recent investigations it has been shown that the specific heat at constant pressure, C_p, is not constant, but that it is approximately 0.65 for 100° F. superheat, and 0.75 for 200° F. superheat. Using these values, it can be calculated that the fuel used to generate saturated steam with superheat must be increased by the following percentages in order to superheat the steam to the various degrees named:

Degree of superheat.	Additional fuel needed.
75°	5 per cent.
100°	7 " "
150°	11 " "
200°	15 " "

Whether, therefore, it is advisable to superheat the steam by direct furnace heat and increase the fuel-consumption or whether it is best to use saturated steam is a problem of finance rather than of engineering. The exception is costless superheat by the waste gases.

Test trials by Professor Schroter, in Belgium, have shown a most decided economy of superheated steam as compared with saturated steam in the same engine, compound-condensing.

The total cylinder-condensation when running with saturated steam was 9 per cent. of the total steam entering the high-pressure cylinder, while with superheated steam the cylinder-condensation was but $4\frac{1}{2}$ per cent. This was at 90 pounds pressure, with superheat at 220° F. The computed economy of superheat for steam, from an average of many trials, was 12 per cent., and for fuel economy an average of 6 per cent.

Saturated steam on leaving the boiler carries water along with it, and to this is added the water of condensation in the pipes and engine-cylinder, amounting to 40 per cent. or more according to the plant

arrangement and to the type of engine. Cylinders provided with jackets heated by saturated steam seldom fulfil their purpose, and then with but small gain; and experience has shown that at times jacketing is a disadvantage. Also, with saturated steam there is the danger of water-hammer in the cylinder and valves. This may be easily prevented where superheated steam is used; for in this case the steam is kept dry until a short time before leaving the cylinder. On account of the much greater volume of superheated steam, a smaller weight is needed to fill the cylinder, or, in other words, the same result is accomplished by the use of a smaller quantity of steam, and this in the case of a condensing-engine means that less water may be circulated in the condenser and consequently smaller pumps may be used. Although the temperature of the exhaust-steam is higher with superheated than with saturated steam, the smaller weight of superheated steam more than compensates for its high temperature, and thus less circulating water is necessary.

Exhaustive comparative tests of saturated and superheated steam for marine purposes have recently been carried out on a steamer called the *James C. Wallace.* This vessel is one of the largest "freighters" on the lakes, and has lately been put into service. She is equipped with two Babcock & Wilcox marine water-tubular boilers with superheaters, and the arrangement is such that the latter may be dispensed with and saturated steam used. The engine is of the quadruple-expansion, vertical direct-acting, jet-condensing type. A comparison based on dry coal shows a net saving in fuel, with superheated steam, amounting to 14.5 per cent. This result represents the combined increased efficiency of the machinery plant. The highest amount of superheat was 91° F.

While differences of opinion may still exist as to what type of superheater is the best, the patient investigation and experiments that have been carried on in Germany have established the fact that superheated steam can be used with locomotives as well as with stationary engines. When the much greater efficiency of superheated over ordinary steam is taken into consideration there seems strong reason to believe that the success already obtained by the stationary engine will be repeated with the locomotive—that materially superior economy in power will be attained by the use of superheated steam, which will in time come into general use for locomotives.

In Europe superheated steam is used with any type of engine, equipped as it may be with poppet-, slide-, piston-, or Corliss valves; and plants built to use saturated steam have later shown the greatest economy with superheated steam. Lubrication of the valves and cylinder is generally accomplished by means of a separate small oil-pump operated by the engine through a ratchet attachment. With compound engines a separate pump is usually provided for each cylinder. The oil used is a high-grade mineral oil with a very high flashing-point, and is extremely thick. With turbines a comparatively smaller pump, operated from the turbine-shaft, is used to supply oil to the steam-inlet and regulating mechanism. A small pump answers the purpose, as a turbine uses only one-sixth to one-tenth of the oil necessary in a reciprocating engine. The stuffing-boxes are made of hard metal; bronze or hard compositions and asbestos or asbestos-graphitic packing are much used.

The mechanical difficulties due to the use of high-temperature superheat will no doubt become a bar to its extensive and continued use. The disintegrating effect upon the best lubricants and the unusual friction-wear of metal and packings at high heat will eventually confine the superheat system to a limited or moderate temperature and to the more economical appliances for heat derived from the waste gases of the chimney—instead of wasting the heat that should go to the steam in the boiler—by the use of fire-chamber devices, or from the losses due to furnace management in separately fired superheaters.

One of the principal wastes in steam-making comes from the heat lost in the chimney, and any saving in this is an economical gain. The temperature of the chimney-gases ranges from 250 to 400 or more degrees Fahrenheit above the temperature of the steam in the boiler, and often much higher than necessary for maintaining proper draught. Every thermal unit rescued from the chimney and added to the steam in the cylinder is a gain that costs nothing for fuel and may add much to the economy in the generation of steam-power, and may be further increased by the decreased consumption of fuel under the boiler.

The saving due to the rescue of heat from the chimney may range from 5 to 14 per cent. of the boiler-fuel, according to the size and economic design of the boiler to meet its required work. Often,

boilers that furnish a scant supply of steam at their limit of pressure may be made to meet the full requirements by the simple addition of a superheating-coil in the chimney-flue.

The value of the specific heat of superheated steam has been the subject of careful experiment, and by a formula from Greisman's experiments the value for superheat is given, for constant pressure, as:

(18) . . . . . $C_p = .00222 t_s - .116$, in which t_s

is the sum of the saturated and the superheat in degrees Fahrenheit. This gives for 100 pounds absolute pressure and 100° F. superheat: $327.6 + 100 = 427.6 \times .00222 = .949 - .116 = .833$; and for the mean specific heat for both saturated and superheated steam at constant pressure. A modification of Greisman's formula, viz.,

$$C_p = \frac{.833 + .48}{2} = .65,$$

has been proposed, which is claimed to be more nearly correct than by using the accepted formula, viz.,

(19) . . . . . . $C_p = .00222 \left(\frac{t_s + t}{2}\right) - .116,$

which for 100 pounds absolute pressure and 100° F. superheat is

$$.00222 \left(\frac{427.6 + 327.6}{2}\right) - .116 = .72.$$

Using this formula for the varying mean specific heat for different pressures and degrees of superheat, the total heat is computed by the formula:

(20) . . . . . $1{,}091.7 + .305(t - 32) + C_p(t_s - t)$

by which the following table (Table XXV) of total heat has been computed for various temperatures of superheated steam and absolute pressures, using the varying values of C_p, as computed from formula (19). For example: for 500° F $C_p = .00222 \left(\frac{500 + 363.3}{2}\right)$ $- .116 = .842$, and $1{,}091.7 + .305(363.3 - 32) + .842(500 - 363.3) =$ 1,307.8, or 1,308, as in the table in the column under 160 and opposite 500° in the first column.

TABLE XXV.—TOTAL HEAT OF SATURATED AND SUPERHEATED STEAM ABOVE 32° F., AT TEMPERATURES IN COLUMN 1, AND ABSOLUTE PRESSURES AT THE HEAD OF THE OTHER COLUMNS.

Temp. Fah.	100	110	120	130	140	150	160	170	180	190	200
380	1,217.0	1,214.8	1,212.6	1,210.5	1,208	1,206	1,205	1,203	1,201	1,199	....
390	1,224.4	1,222.2	1,220.0	1,218.0	1,216	1,214	1,212	1,210	1,208	1,206	1,204
400	1,232.0	1,229.8	1,227.6	1,225.5	1,223	1,221	1,219	1,217	1,215	1,214	1,212
410	1,239.9	1,237.6	1,235.5	1,233.4	1,231	1,229	1,227	1,225	1,223	1,222	1,220
420	1,247.9	1,245.7	1,243.5	1,241.4	1,239	1,237	1,235	1,233	1,231	1,230	1,228
430	1,256.3	1,253.9	1,251.8	1,249.7	1,247	1,246	1,244	1,242	1,240	1,238	1,236
440	1,264.6	1,262.4	1,260.3	1,258.2	1,256	1,254	1,252	1,250	1,248	1,246	1,244
450	1,273.3	1,271.2	1,269.0	1,266 0	1,265	1,263	1,262	1,259	1,257	1,255	1,253
460	1,282.3	1,280.1	1,277.9	1,276.0	1,273	1,271	1,269	1,267	1,266	1,264	1,262
470	1,291.4	1,289.3	1,287.1	1,285.0	1,283	1,281	1,279	1,277	1,275	1,273	1,271
480	1,300.8	1,298.7	1,296.5	1,294.0	1,292	1,290	1,288	1,286	1,285	1,283	1,281
490	1,310.4	1,308.3	1,306.1	1,304.0	1,302	1,300	1,298	1,296	1,294	1,292	1,290
500	1,320.2	1,318.1	1,315.9	1,314.0	1,312	1,310	1,308	1,306	1,304	1,302	1,300
510	1,330.2	1,328.2	1,326.0	1,324.0	1,322	1,320	1,318	1,316	1,314	1,312	1,310
520	1,340.6	1,338.4	1,336.2	1,334.0	1,332	1,330	1,328	1,326	1,324	1,322	1,320
530	1,351.0	1,349.0	1,346.0	1,345.0	1,342	1,341	1,339	1,337	1,334	1,333	1,331
540	1,361.8	1,359.0	1,357.0	1,355.0	1,353	1,351	1,349	1,347	1,346	1,344	1,342
550	1,372.9	1,371.0	1,368.0	1,366.0	1,364	1,362	1,360	1,358	1,357	1,355	1,353

SUPERHEATERS AND THEIR CONSTRUCTION

The most simple form of a superheater is a coil of ordinary steam-pipe, extra heavy for wear, bent into a circular shape or made up with return-bends, and set in the chimney-flue, or, if needed, over the fire in a separately fired furnace. The bent pipe-coils are much in use for obtaining high temperatures from superheated steam in japanning-ovens and in vulcanizing processes for hard-rubber goods. In this manner, by circulating superheated steam in pipe-coils, an oven temperature of 275° may be readily obtained.

Any form of superheating-coil placed in the chimney-flues of boilers having ample heating-surfaces for their required output of steam, and in which the economy from chimney-waste has been kept within reasonable limits above the steam temperature, and from which any degree of superheat can be obtained, is a saving without cost.

In the vast number of so-called economic types of boilers and their setting, a saving of 100° F. from the chimney-gases by superheat in the steam will make a decided saving in fuel and in boilers having large heat-waste from overwork; a considerable increase in power may be

obtained at the first cost of a simple coil of pipe and its setting in the chimney-flue.

In Fig. 126 is shown a group of tubes ready for setting in a flue or separate furnace; it consists of pairs of cast-iron pipes with solid bends, arranged as shown, and filled with iron-wire coils that produce intercirculation of the steam for quick-heat action by convection.

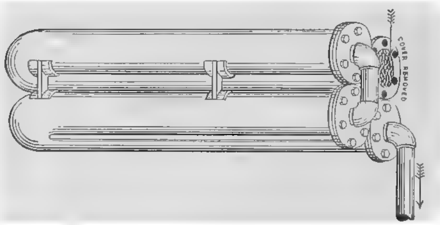


Fig. 126.—Bulkley superheater.

In Fig. 127 is shown the arrangement of the heater in a special furnace—H. W. Bulkley type. Steam may be heated to 800° in these superheaters, the temperature of which is shown by a pyrometer in the exit-pipe.

In Fig. 128 is shown the Metesser type of superheater-coil, which consists of steel tubes bent into the form shown, with their ends expanded into a thick steel plate with a steel or cast-iron backing divided

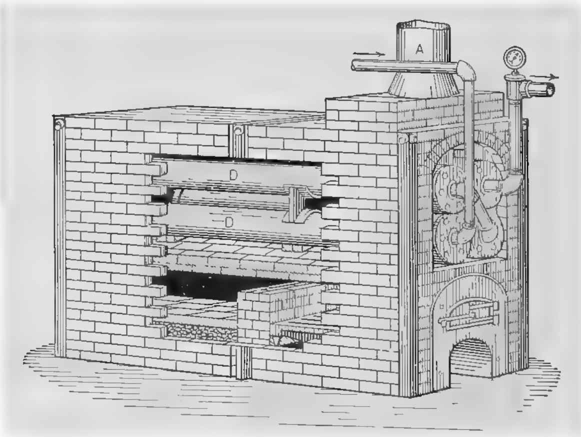


Fig. 127.—Bulkley superheater in brick setting.

into two compartments. The two parts are bolted together with corrugated copper gaskets between the flanges. The tube-section may be hung in a flue-chamber or placed across the rear end of a water-tube boiler, in which latter case the superheater is placed in and securely bolted at the tube-sheet end to an iron frame, which is firmly anchored in the boiler-wall, while the free ends of the tubes enter a

recess in the opposite wall and are prevented from sagging by supports placed between them. By this arrangement no joints of any kind are in the hot gases.

In Fig. 129 are represented the pipe connections for a return-bend superheater placed across and over the tubes of a duplex water-tube

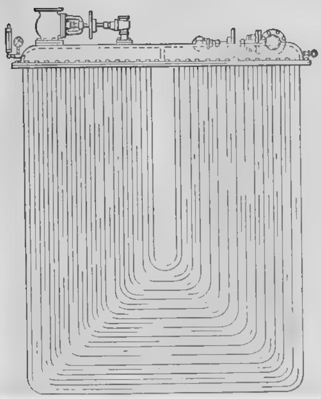


FIG. 128.—Metesser superheater.

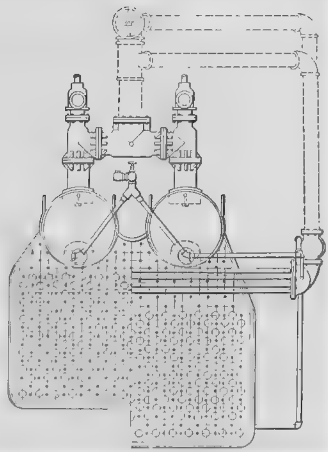


FIG. 129.—Superheater in rear end of boiler.

boiler. Provision is also made for flooding the superheater with water from the boiler when the engine is not running.

In Fig. 130 is shown a cluster-tube superheater set in a flue-chamber, in which steel tubes of suitable size, bent into U-shape, are flanged on, or screwed to the headers with right-and-left couplings in rows and in number of tubes to contain the required fire-surface.

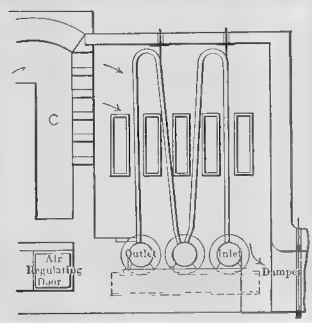


FIG. 130.—Flue-chamber superheater.

In Fig. 131 is shown a superheater made with pipe and return-bend with rib-flanges pushed over pipes for extending the heating-surface and for protection from the direct contact of the gases. It is placed vertically in the rear chamber of a horizontal tubular boiler. This position of the superheater does not

contribute to the efficiency of the boiler, although it may be an effective superheater. The principle of abstracting heat that should pass through the tubes of the boiler is of doubtful economy.

In Fig. 132 is shown a return-bend coil with rib-flanges placed in the rear fire-chamber of a marine boiler. This form is also applicable to the smoke-boxes of locomotives, and in the various ways and designs in which it may be applied adds largely to counteracting

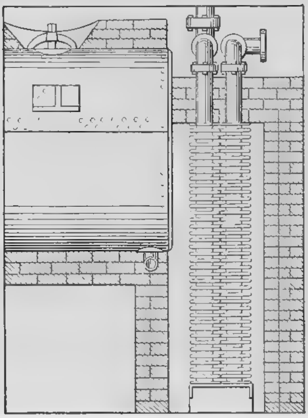


Fig. 131.—Rear-chamber superheater.

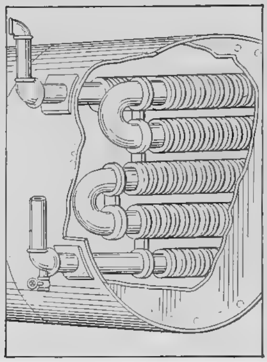


Fig. 132.—Superheater at rear end of marine boiler.

the effect of cylinder-exposure; as applied to a marine boiler the separate fire, with its inconvenient conditions, is avoided.

In Fig. 133 is shown a sectional view of a separately fired superheater and furnace. In the bridge-wall of the furnace there is an air-inlet for tempering the heat of the furnace before it reaches the superheater-coil. In this way the amount of superheat is controlled and overheating of coil prevented when the engine is not running.

In Fig. 134 are shown the details of construction of the Schwoerer superheater, much in use in Europe. The inside ribs and outside flanges, cast in and on the pipe, with the method of connecting them with the return-bends, are shown. The tubes may be disposed either vertically or horizontally according to the place where they are to be located, and may be installed either in the uptake from the boiler, in the boiler-furnace, or in a setting to be separately fired. The tubes are of cast iron, with transverse flanges on the outside to take

up heat from the gases, and longitudinal ribs on the inside to give up heat to the steam. These flanges and ribs are necessary on ac-

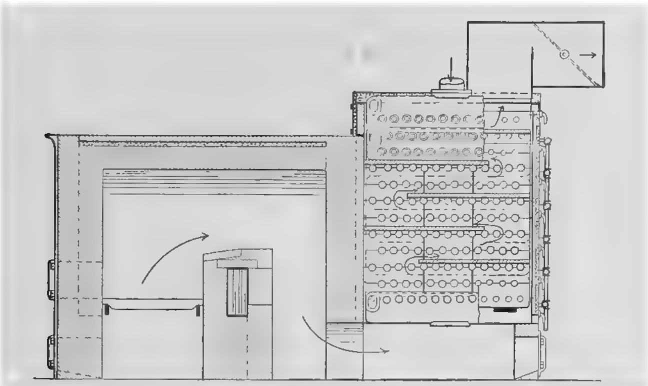


FIG. 133.—Superheater with separate furnace.

count of the poor conducting power of the superheated steam, which makes it necessary to have large surfaces for the transfer of heat.

This cast-iron construction gives a large mass of hot metal which serves as a magazine for heat and acts to hold at an even temperature the superheated steam which is delivered. In making joints between the tubes and the connecting-bends, strong flanges are used connected by heavy bolts. Each flange is turned with a circular groove

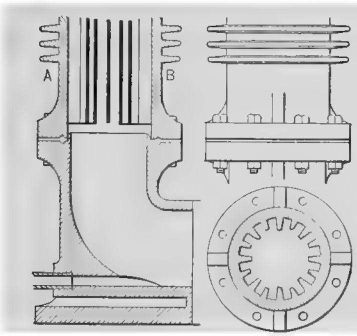


FIG. 134.—Schwoerer superheater.

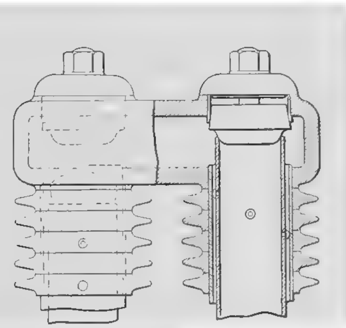


FIG. 135.—Foster superheater.

of triangular section, and into these grooves are placed steel rings of corresponding form. These steel rings, strongly compressed between the flanges, give an iron-to-iron joint which is sure to remain tight.

In Fig. 135 are shown some of the details of the Foster superheater,

illustrating the ends of the elements connected by a return-header. The elements consist of concentric, seamless, drawn-steel tubing protected by cast-iron rings, shrunk on. The inner tubes are closed to the steam, which is thus forced through thin annular spaces and rapidly superheated.

In Fig. 136 is illustrated a longitudinal section of the Babcock & Wilcox water-tube boiler, with the location of the superheater, and in Fig. 137 a cross-section of the superheater and its steam-connections.

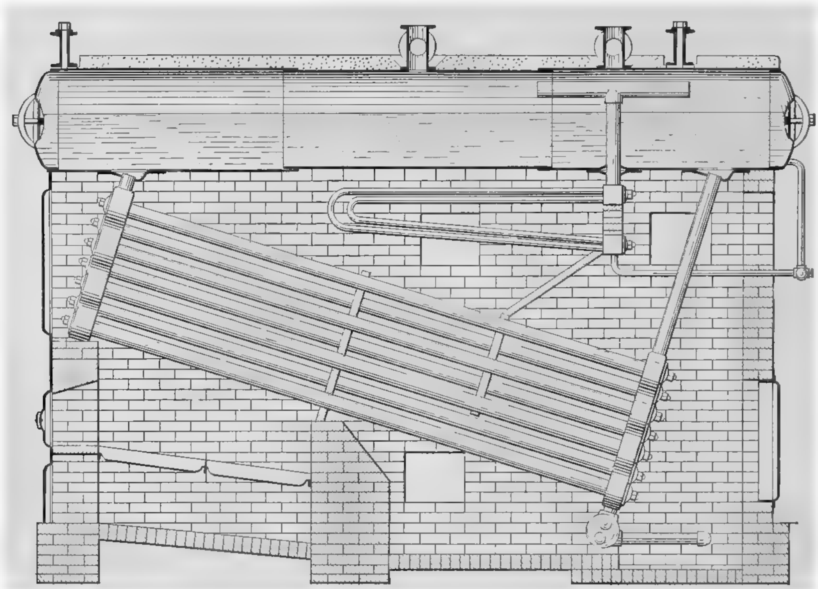


FIG. 136.—Longitudinal section of Babcock & Wilcox boiler and superheater.

This superheater is not subject to the immediate action of the fire, as the furnace-gases must first pass through the front part of the boiler, which comprises a considerable heating-surface. Assuming the boiler to be in regular work and the firing even, no great fluctuations in temperature can take place where the superheater is fixed. Moreover, it is readily accessible for examination and for the renewal of tubes.

There are no flanged joints; all the tube-joints are expanded and freedom for expansion is provided by the tubes being free at one end, and by the manifolds not being rigidly connected with each other.

Prevention against overheating during steam-raising is insured by the arrangement for flooding with boiler-water and using the superheater as part of the boiler heating-surface while steam is being raised or when it is desired to use saturated steam.

As will be seen, the tubes are bent into a U-shape and connected at both ends with manifolds, one of which receives the natural steam from the boiler, the other collecting the superheated steam after it has traversed the superheater-tubes and delivering it to the valve placed above the boiler.

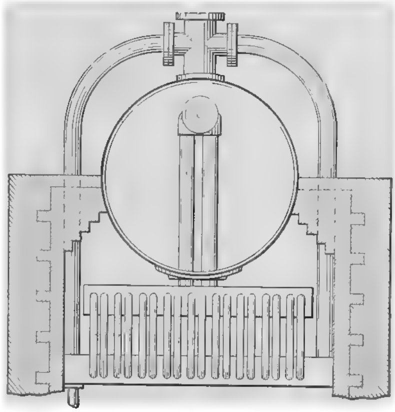


FIG. 137.—Cross-section Babcock & Wilcox superheater.

The flooding arrangement consists merely of a connection with the water-space of the boiler-drum and a three-way cock, by which the water enters the lower manifold and fills the superheater to the boiler water-level. Any steam formed in the superheater-tubes is returned to the boiler-drum through the collecting-pipe, which, when the superheater is at work, conveys saturated steam into the upper manifold through the heating-tubes, and from the lower manifold, by two tubes outside of the drum, to the fitting at the top of the boiler.

In Fig. 138 is shown a section of the W. Schmidt superheater.

The management of superheaters is of interest, and we append a short description of the Schmidt superheater, which is also applicable to other types or models.

Superheating steam under the Schmidt system may be effected in one of two ways, either by placing the superheater in a chamber between the boilers and the main flue—this being known as the flue-fired superheater—and using a portion of the hot gases direct from the boiler-flue, or by having an independent, direct-fired superheater through which the saturated steam from the boiler, or battery of boilers, is made to pass before reaching the engine. The illustration (Fig. 138) represents a section of the setting and the construction of the apparatus.

The saturated steam enters through the valve at the top, and, having been dried in the upper half of the apparatus, is led through suitable passages to the bottom tubes, cooling them from the inside and so protecting them from deterioration. It then flows in the same direction as the flue-gases, taking up heat from them on the way. The higher the temperature of the steam, the less that of the gases, and the steam leaves the superheater when it is hottest. The gases leave the superheating-coils at about 900° F., and pass on to the drying-coils, whence they enter the main flue at a temperature of about 460° F. The heat of the gases is thus utilized to the highest possible extent, while at the same time the tubes are sufficiently protected from excessive heat.

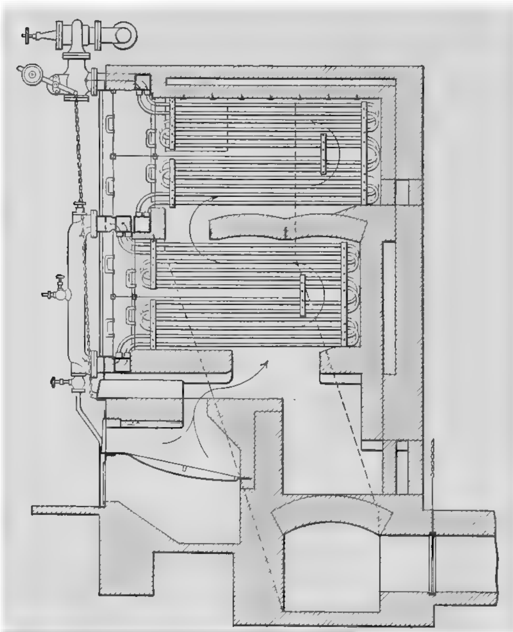


Fig. 138.—Schmidt superheater.

The superheater consists of a number of coils of equal size and dimensions, the ends of which are fixed to cast-iron junction-boxes. All boxes are placed outside the chamber, thus avoiding contact with the flue-gases, and are easily accessible even when the superheater is in service. Each coil can be taken out separately and a new one put in without removing the others or dismantling the plant. If one coil becomes defective, it need not be replaced at once; the ends can be stopped with blank flanges in a few minutes, and the tube replaced when convenient.

All the water produced by condensation while the superheater is idle collects in the bottom junction-box and escapes through the drain-cock.

The outside of the coils should be cleaned at intervals according to the nature of the fuel employed. The cleaning is effected with a

jet of steam in the usual way. A steel mercury thermometer, scaled to 900° F., is fitted where the superheated steam enters the main steam-pipe, and has a red mark to indicate the maximum temperature. When this mark is passed, an electric bell rings as long as the maximum temperature is exceeded. A thermometer-pocket is also provided, in which a glass thermometer can be placed for checking the steel mercury thermometer.

When starting, the superheater should be warmed with steam while the engine is warming up, and care should be taken to leave the drain-cock open until the engine has actually started. After the engine has run for a few minutes the cock should be closed and the superheater brought into operation.

If by chance the steam should be suddenly cut off, the air-door underneath the lowest junction-box should be opened to enable cool air to enter and protect the tubes from the fire and from radiation of heat from the walls. This door is automatically worked by a valve kept closed by a weight attached to an outside lever. A chain connects the weight with the air-door and tends to keep it open. As soon as the steam begins to flow it presses the valve downward, and as the valve falls the weight is lifted and closes the door.

Where an engine works continuously, and is in charge of a competent stoker, this apparatus is unnecessary, and can be put out of action by merely disconnecting the chain, but in cases of irregular working, and especially when the engine is liable to be stopped suddenly without the stoker's knowledge, the arrangement is of the greatest importance.

As a general rule the stoking of the superheater should cease about three-quarters of an hour before the engine is to be stopped, so that when the superheater is put out of action the fire will be out and the bricks cooled down to some extent. During this period the temperature gradually decreases, but the stored heat is sufficient to keep the steam at the required temperature until the engine stops.

This apparatus is manufactured by the Providence Engineering Works.

The conditions for enabling the use of high-pressure steam and high superheat with safety may be summarized as follows:

1. A large factor of safety.

2. Steam and water capacity sufficient to care for sudden fluctuations of load.

3. A proper arrangement of heating-surface to thoroughly absorb the heat of the gases.

4. The absence of any stayed surfaces.

5. Straight tubes, so that they can be cleaned easily, and so that one can see through them and know that they are clean.

6. Sectional construction to insure safety and ease of repair.

7. Wrought-steel construction throughout.

8. Ample surface to disengage the steam easily so as to avoid priming or a fluctuating water-level.

9. Expansion and contraction properly provided for.

10. And last, but not least, a perfect and positive circulation of the water in the boiler.

Added to the above it is imperative, on the score of economy, that the soot can be easily removed from the heating-surface while the boiler is in operation, preferably by means of air- or steam-jets.

In Fig. 139 is illustrated a high-pressure boiler of the Babcock & Wilcox type with a double bend superheater coil set in the upper

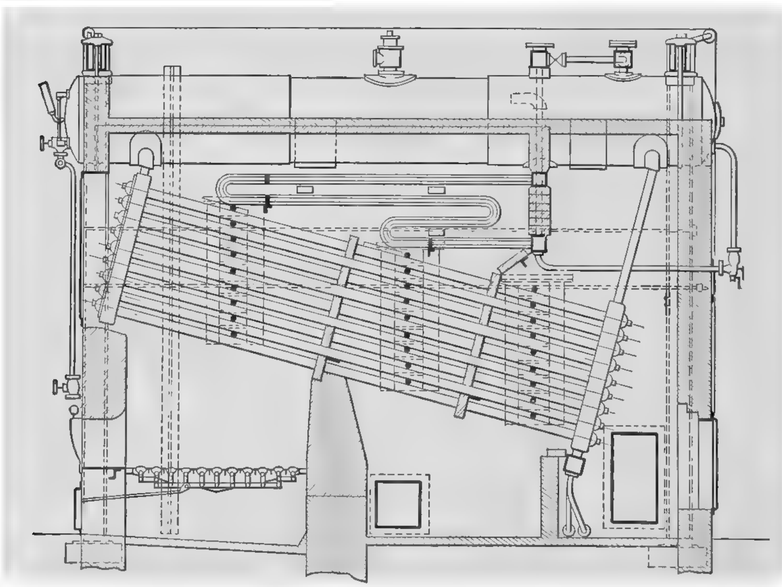


FIG. 139.—Boiler and superheater for 200 pounds pressure and 150° F. superheat. Babcock & Wilcox type.

chamber with its connections to the boiler shell, arranged as before described.

In conclusion, it might be well to further emphasize the advantages of moderate superheat of, say, 100 to 150° F.

In most plants, if properly piped and protected, this amount of superheat will not only avoid condensation in the steam-mains, but will practically eliminate the condensation losses in the high-pressure cylinders of the engine, and this alone will show an actual saving, varying from 10 to 25 per cent. according to the class of engine and its condition. This, coupled to the saving effected by the use of high-pressure steam, and to boilers that can be cleaned and kept up to their efficiency while at work, is of such importance in considering the cost of operation that no thinking user of steam can afford to disregard it.

THE MEASUREMENT OF STEAM-CONSUMPTION

The quantity and value of steam sold for heating and power purposes to other parties, and which must be delivered through

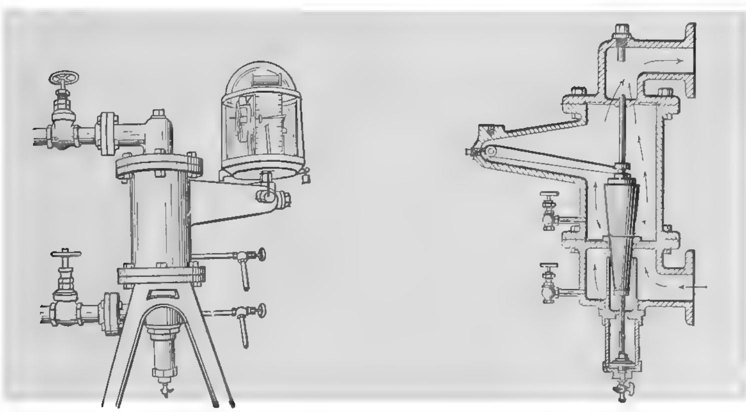


Fig. 140.—Steam-meter. **Fig. 141.—Section of the steam-meter.**

pipes, may be measured with fair accuracy, even when its use is variable or intermittent.

In Figs. 140 and 141 we illustrate the automatic recording steam-meter of Mr. G. C. St. John, of New York City, which makes a record on a chart moved by clock-work that shows the horse-power that is

being used at all times, and the aggregate per day or month. Many hundreds are in use in the New York steam service and throughout the country. The lifting of a conical valve by differential pressure allows the required quantity of steam to pass through the annular area, which is the measure under the initial pressure. The valve-lift is recorded on a strip of paper moved by a clock; the mean of the record-curves being the measure for the time. The marking-hand is moved by a lever from the conical valve and by a small transfer-shaft through the projecting hollow arm from the cylinder. The small chamber at the bottom is a dash-pot filled with water, which keeps the valve from chattering.

The sale of steam for power and for heating purposes, in manufacturing districts, is generally made in horse-power units, and when supplied to engines only, the indicated horse-power of the engine is the usual measure of the steam-supply, unless the waste by condensation in long pipe-lines may require an additional allowance.

For heating purposes the unit for the price may be the same as for power; but the method used for obtaining the unit, when meter measurement is out of the question, is often a matter of controversy from personal differences in regard to space, and exposure of heated areas and their required temperature. The only reliable method of measurement that is available is derived from the weight of water drained from the heating-pipes and its weight as steam at the pressure in the supply-pipe.

The horse-power in ordinary slide-valve engines varies somewhat from 20 pounds per horse-power hour, which may be taken as a fair average for indicating the amount of steam used for heating purposes.

CHAPTER XII

ADIABATIC EXPANSION OF STEAM

In adiabatic expansion without loss or gain in heat from outside source or from the walls of a cylinder, the terms of expansion are:

$$P_1V_1{}^y = P_2V_2{}^y; \text{ then } \frac{P_2}{P_1} = \frac{V_1{}^y}{V_2{}^y} \text{ and } \frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{y}}; \text{ also,}$$

$\left(\frac{V_1}{V_2}\right)^{y-1} = \left(\frac{P_2}{P_1}\right)^{\frac{y-1}{y}}$, and is also applicable to the relation of pressures and volumes to temperatures, and

$\left(\frac{V_1}{V_2}\right)^{y-1} = \frac{T_2}{T_1}$ and $\left(\frac{P_2}{P_1}\right)^{\frac{y-1}{y}} = \frac{T_2}{T_1}$; also, $\left(\frac{1}{r}\right)^{y-1} = \frac{T_2}{T_1}$, in which P = pressure in pounds absolute or per square foot, V = volumes in cubic feet, r = ratio of volumes. T = temperatures before and after expansion.

The ratio exponent in any equation for the adiabatic expansion of steam is variable as discussed by leading authorities, and as a sensibly perfect gas or superheated steam is given as 1.35; and for saturated steam varying from 1.3 to 1.111, by different authors, or $\frac{10}{9}$, as adopted by Rankine for the consideration of the actual condition of steam behind a moving piston. Probably there is no exact empirical value for ^{y} for the varying influences in the make-up and time of expansion in a steam-cylinder. Professor Wood gives the specific heat of steam at constant pressure, $C_p = 373.44$, and for constant volume, $C_v = 290.16$, and $\frac{C_p}{C_v} = \frac{373.44}{290.16} = 1.2869 = {}^y$ for their ratio in foot-pound values.

Professor Zeuner found that the value of ^{y} depended upon the specific volume of the steam for the same initial condition at from 1 to 4 atmospheres, and that the true value may be represented by the empirical formula:

(21) . . $^y = 1.035 + 0.100x$, x being derived from the formula used for computing Column 8, Table XXI, which is proportional for the expansion of steam from 160 absolute to other lower pressures.

Then for any degree of expansion between limited pressures the exponent ^{y} will be $1.035+0.100x$, as computed by formula (21).

For expansion from 160 to 15 pounds absolute x may be taken as $.865 \times 0.100 = .0865 + 1.035 = 1.1215 = {}^y$, and $\frac{1}{y} = .891$. Then $\left(\frac{P_1}{P_2}\right)^{\frac{1}{y}} = \frac{160}{15} = 10.66$ log. $1.027787 \times .891 = 0.915758217 =$ index 8.237, the volume of saturated steam expanded from 160 to 15 pounds absolute.

The water of condensation by the expansion of steam in this ex-

Table XXVI.—Real Cut-Off, Corresponding to the Apparent Cut-Off, for Different Fractions of Clearance.

Apparent cut-off.	Real cut-off for fraction of clearance of								
	.01	.02	.03	.04	.05	.06	.07	.08	.09
.08	.090	.098	.106	.115	.124	.132	.140	.148	.155
.10	.109	.118	.126	.135	.143	.151	.159	.167	.174
.12	.129	.138	.146	.154	.162	.170	.177	.185	.192
.14	.149	.158	.165	.174	.181	.189	.196	.204	.211
.16	.169	.177	.184	.193	.200	.207	.215	.222	.229
.18	.189	.197	.204	.212	.219	226	.233	.240	.247
.20	.208	.216	.223	.231	.238	.245	.252	.259	.266
.22	.228	.236	.243	.251	.257	.264	.271	.277	.284
.24	.248	.256	.262	.270	.276	.283	.290	.296	.303
.26	.268	.275	.281	.289	.295	.302	.308	.315	.321
.28	.288	.295	.301	.308	.314	321	.327	.333	.339
.30	.307	.314	.320	.327	.333	340	.346	.352	.358
.32	.327	.334	.340	.346	.352	.359	.364	.370	.376
.34	.347	.354	.359	.366	.371	.378	.383	.389	.395
.36	.367	.373	.378	.385	.390	.396	.402	.407	.413
.38	.387	.393	.398	.404	.409	.415	.420	.425	.431
.40	.406	.412	.417	.423	.429	.434	.439	.444	.450
.42	.426	.432	.437	.442	.448	.453	.458	.462	.468
.44	.446	.452	.456	.462	.467	.472	.477	.481	.486
.46	.465	.471	.475	.481	.486	.490	.495	.500	.504
.48	.485	.491	.495	.500	.505	.509	.514	.518	.522
.50	.505	.510	.514	.519	.524	.528	.533	.537	.541
.52	.525	.530	.534	.538	.543	.547	.551	.555	.559
.54	.545	.550	.554	.558	.562	.566	.570	.574	.578
.56	.564	.569	.573	.577	.581	.585	.589	.593	.596
.58	.584	.589	.593	.596	.600	.604	.607	.611	.614
.60	.604	.608	.612	.615	.619	.623	.626	.630	.633
.62	.624	.628	.632	.634	.638	.642	.645	.648	.651

ample is: 1 − .865, or $13\frac{1}{2}$ per cent.; and from the ratio of volumes at the above pressures $\frac{25.85}{2.80}=9.232$, and $\frac{8.237}{9.232}=.111$ per cent. condensation from the effect of expansion alone.

In practice, the cooling effect of the cylinder-walls increases the percentage of condensation, which may reach 25 per cent. in slow-running engines. Thus the speed of piston is one of the claims for economy for high-speed engines.

The values of the real cut-off in the table are derived from the equation $\frac{r+c}{1+c}$, in which r = the ratio of the apparent cut-off, and c = the percentage of clearance. For example: for .30 cut-off with 7 per cent. clearance, .30 + .07 = .37, and $\frac{.37}{1.07}=.3457$, or .346, as in the table.

The formulas for mean forward pressure of expanding steam, as given by authorities who have critically investigated this subject, vary somewhat from the results given by the hyperbolic formula, which is now accepted as more nearly meeting the exact conditions of steam-engine practice. Rankine's formula $P_1\left(\frac{10}{r}-\frac{9}{r^{\frac{10}{9}}}\right)$ seems to give for mean forward pressure about 2 per cent. in excess of the hyperbolic formula.

In Table XXVII are given the decimal multipliers for the mean forward absolute pressure, with the apparent or nominal cut-off and clearance of from 1 to 10 per cent. of the stroke of the piston.

For obtaining the multiplier for mean forward pressure for any absolute pressure, as shown in the table, from the hyperbolic formula, we have

(22) . . $\left(\frac{1+\text{hy. log. } R}{R}\times 1+c\right)-c$, in which R = the ratio of expansion, or $\frac{1}{\text{real cut-off}}$, as in Table XXVI. Then by substituting the values for, say, .30 cut-off and 7 per cent. clearance, we have

$\frac{.37}{1.07}=.346$, and $\frac{1}{.346}=2.89$, the ratio R of expansion, the hyp. log. of which is $1.0613+1=\frac{2.0613}{2.89}=.713\times 1.07=.762-.07=.692$, as in the table.

TABLE XXVII.—MEAN FORWARD PRESSURE FROM ABSOLUTE INITIAL PRESSURE, WITH ACTUAL CLEARANCE DUE TO THE NOMINAL CUT-OFF.

	Cut-off, nominal.	Clearance, per cent. of stroke.									
		.01	.02	.03	.04	.05	.06	.07	.08	.09	10
$\frac{1}{10}$	.10	.344	.357	.369	.381	.392	.402	.413	.423	.432	.441
$\frac{1}{9}$	.111	.368	.380	.391	.402	.413	.423	.433	.442	.451	.459
$\frac{1}{8}$	.125	.397	.403	.418	.429	.439	.448	.457	.467	.474	.482
$\frac{1}{7}$	.143	.432	.442	.452	.460	.470	.479	.488	.495	.503	.510
$\frac{1}{6}$	.167	.475	.484	.493	.501	.509	.517	.524	.531	.538	.545
$\frac{1}{5}$	.20	.530	.538	.545	.552	.559	.566	.572	.578	.584	.590
$\frac{1}{4}$	.25	.603	.609	.615	.621	.626	.631	.637	.641	.646	.650
$\frac{3}{10}$	.30	.666	.671	.675	.679	.685	.688	.692	.697	.701	.705
$\frac{1}{3}$	.333	.704	.708	.712	.716	.719	.722	.726	.731	.734	.737
$\frac{2}{5}$	.40	.769	.772	.776	.778	.781	.784	.787	.789	.791	.794
$\frac{1}{2}$	.50	.848	.850	.852	.854	.856	.858	.860	.861	.863	.864
$\frac{5}{8}$	.625	.919	.920	.921	.923	.925	.925	.926	.927	.927	.928
$\frac{3}{4}$	.75	.967	.967	.968	.968	.969	.969	.969	.970	.970	.970

From the mean absolute forward pressure the actual back pressure must be subtracted for obtaining the mean effective pressure due to the piston-stroke. If the exhaust is directly to the atmosphere, the atmospheric pressure, plus the back pressure due to the friction in pushing the steam before the piston and through the exhaust-pipe, will be the total back pressure.

The variation of the atmospheric pressure may be from 14 to 15 pounds, according to the barometric pressure, and the back pressure from 1 to 3 pounds more, depending upon the frictional conditions in the exhaust.

Terminal pressure is due to the stroke as 1 divided by the ratio of expansion, or 1 divided by the real cut-off, as found in Table XXVI, which gives the ratio of expansion; and for the last example

$\frac{1}{.346}=2.89$, and $\frac{1}{2.89}=.346$, the multiplier for the initial absolute pressure.

For example: for 100 pounds initial absolute pressure, $\frac{3}{10}$ cut-off with 7 per cent. clearance the mean forward pressure per Table XXVII is .692; the real cut-off per Table XXVI is .346, which is also the ratio for the terminal absolute pressure in Table XXVIII.

Then, for example, $100\times.692\times.346=23.94$, absolute, and $23.94-14.7=9.2$, the terminal gauge pressure.

TABLE XXVIII.—TERMINAL ABSOLUTE PRESSURE DUE TO THE ABSOLUTE FORWARD PRESSURES AND CLEARANCE, AS GIVEN IN TABLE XXVII. ABSOLUTE INITIAL PRESSURE × MEAN FORWARD PRESSURE × TERMINAL PRESSURE DUE TO CUT-OFF TERMINAL ABSOLUTE PRESSURE.

Cut-off.	Terminal absolute pressure for fraction of clearance of								
	.01	.02	.03	.04	.05	.06	.07	.08	.09
.08	.09	.098	.106	.115	.124	.132	.140	.148	.155
.10	.109	.118	.126	.135	.143	.151	.159	.167	.174
.12	.129	.138	.146	.154	.162	.170	.177	.185	.192
.14	.149	.158	.165	.174	.181	.189	.196	.204	.211
.16	.169	.177	.184	.193	.200	.207	.215	.222	.229
.18	.189	.197	.204	.212	.219	.226	.233	.240	.247
.20	.208	.216	.223	.231	.238	.245	.252	.259	.266
.22	.228	.236	.243	.251	.257	.264	.271	.277	.284
.24	.248	.256	.262	.270	.276	.283	.290	.296	.303
.26	.268	.275	.281	.289	.295	.302	.308	.315	.321
.28	.288	.295	.301	.308	.314	.321	.327	.333	.339
.30	.307	.314	.320	.327	.333	.340	.346	.352	.358
.32	.327	.334	.340	.346	.352	.359	.364	.370	.376
.34	.347	.354	.359	.366	.371	.378	.383	.389	.395
.36	.367	.373	.378	.385	.390	.396	.402	.407	.413
.38	.387	.393	.398	.402	.409	.415	.420	.425	.431
.40	.406	.412	.417	.423	.429	.434	.439	.444	.450
.42	.426	.432	.437	.442	.448	.453	.458	.462	.468
.44	.446	.452	.456	.462	.467	.472	.477	.481	.486
.46	.465	.471	.475	.481	.486	.490	.495	.500	.504
.48	.485	.491	.495	.500	.505	.509	.514	.518	.522
.50	.505	.510	.514	.519	.524	.528	.533	.537	.541

The available heat in steam for power is essentially the sensible heat, that can create energy by expansion from any initial temperature and pressure to some lower temperature and pressure. The total available energy from 85 pounds gauge, 100 absolute, is: $327.6-212=115.6°$ F.$\times 778=89{,}936$ foot-pounds, or nearly $2\frac{3}{4}$ horse-power per pound of steam—less friction, condensation, radiation, and leakage—in any mechanical device for utilizing its energy.

The available heat of the exhaust (latent heat) is between its temperature at atmospheric pressure and the temperature of the water after condensation, say 150° F.; then $212-150=62°\times 778=$ 48,226 foot-pounds, or nearly $1\frac{1}{2}$ horse-power per pound of steam. Then $4\frac{1}{4}$ horse-power is the greatest available energy that can be obtained from 1 pound of steam, at 85 pounds gauge-pressure, by expansion and condensation. The practical operation of conversion is variable, and much less than the theoretical deduction.

Steam, when suddenly expanded, as in a cylinder, suffers condensation by a small percentage, and the latent heat thus liberated is added to the remaining uncondensed steam. The amount, independent of the condensation by contact with the cylinder-walls, is shown in Column x, Table XXI, and by the formula from which that column was computed.

When steam is compressed, as in cylinder-compression, the contrary effect is produced; the heat generated by compression is added to the steam and it becomes superheated. The economy of high-pressure steam has become more evident as discussion and experimentation have greatly advanced its possibilities during the past two decades; so that practically, from theoretical deduction, the total heat that is available for power advances with the initial pressure at a greater rate than the latent heat, as shown in the total latent-heat columns of the table of properties of saturated steam (Table XX). For instance, the difference at 100 pounds absolute is 298.9 heat-units, and at 200 pounds is 355 heat-units, or over 18 per cent. in available heat-units.

The interchangeable heat effect from steam in contact with the surface of the cylinder-walls is made evident by the well-known difference in temperature of the steam at entrance and at exhaust. From the initial temperature during the period of admission, every part of the cylinder-walls in contact with the incoming steam—cylinder-head, piston, piston-rod, and passages—receives heat from the initial temperature of the steam; during which time condensation takes place upon their surface, and the latent heat liberated by condensation is absorbed by the cylinder-walls, which have become cooled by the lower temperature of the previous exhaust. During the period of expansion the temperature of the steam falls, so that at near the terminal it is below that of the walls that received heat by the previous admission, and reëvaporation takes place; thus latent heat is liberated by transfer during admission and absorbed by reëvaporation during expansion and the exhaust. The balance is small with high-speed pistons, yet in no case is there an absolute balance obtained, except at the expense of steam-jacket addition of heat to counteract radiation and air-convection.

ECONOMY OF THE SIMPLE HIGH-SPEED ENGINE

During the past two decades the economy of steam-engine design and its use of steam has been a fruitful source of discussion and experiment, resulting in reducing the comparative length of stroke, in increase of speed, and in the adoption of more perfect and automatic valve-motion and economic cut-off. These points are still variable in the designs of engine-builders, but are verging toward a uniform ideal. Both theory and practice now show that increased economy in the use of steam is found from increased pressure to certain limits for single-expansion, from the fact that there is more heat in higher-pressure steam available for doing work, in proportion to the amount of heat required to generate the steam, than in the case of low-pressure steam with its proportionate loss in doing useless work; nor is there any gain in extremely high pressure for single-cylinder engines, because of the loss from condensation due to the extreme range of temperature that would result from extreme pressures.

In a simple non-condensing, high-speed engine the limit of economic pressure may be at 115 pounds gauge-pressure, and in simple condensing-engines there is little advantage with steam above 90 pounds.

In the long-stroke system, with single valves and long steam-passages, it is certain there is a large amount of cooling-surface that the steam is in contact with, while entering the cylinder, that is cooled by the lower temperature of the exhaust through the same passages.

When the valves are close to the ends of the cylinder, as in Corliss and other types of four-valve engines, the surfaces of the ports and port-passages are reduced to only a trifle greater than due to the thickness of the cylinder-wall, which leaves only the cylinder-heads and piston with the small section of cylinder-wall to condense the incoming steam. In this type of engine there is economy.

In the single-valve automatic engine we have a condition that, while simple and compact, loses a little in economy, because of the long steam-passages, and from the fact that the cool exhaust-steam must pass through the same passages and the same valve from which the live steam enters. With engines having the single valve there is

not as good steam-distribution as when two or four valves are used, with double eccentrics. In the single-valve engines the characteristics are that the earlier in the stroke the steam is cut off, the greater will be the compression, and that at very early cut-off the compression will become excessive; thus we see that it is not possible to operate the engine with an early cut-off and realize the full benefit of expansion. The clearance-surfaces are, however, warmed up by the compression, which is a benefit in its way.

There is no definite rule which would tell how far to carry the expansion in any particular case, although it is generally considered that the best results are obtained in the case of non-condensing engines when cutting off at about one-third stroke. With a simple condensing-engine the best results are usually obtained when cutting off at from one-sixth to one-fourth stroke. In compound engines varying degrees of expansion are used, the point of cut-off in the high-pressure cylinder usually being adjusted to give from 12 to 20 expansions in both cylinders.

When an engine is overloaded it is useless to expect to operate with economy, and the same will apply when the engine is too large to do the work, because the point of cut-off will come in one case too late, and in the other too early, for the economical use of steam. If the cut-off occurs too early there will be an increased loss from cylinder-condensation, and if too late, the expansion of the steam will not be carried out as far as it should be. It has been found that when an engine is running under these conditions it is better, if possible, to change the steam-pressure, or else the speed of the engine, so as to allow the cut-off to occur at a point more nearly at its correct position.

Quoting from a series of tests which we have before us and which were made upon Corliss engines of medium size, we find that the amount of condensation and leakage, taken together up to the point of cut-off, was 60 per cent. of the steam consumed when the cut-off was at 5 per cent. of the stroke, 45 per cent. with the cut-off at 10 per cent., 35 per cent. with the cut-off at 15 per cent., 30 per cent. with the cut-off at 20 per cent., 20 per cent. with the cut- off at 30 per cent., and 15 per cent. with the cut-off at 40 per cent.

It will be seen from the above that the percentage of loss decreases as the point of cut-off grows later, and the later cut-off may cause as

much condensation, and even more, as with an early cut-off, owing to the large quantity of steam used when the cut-off is late in the stroke. It is evident that it is better to operate an engine with too heavy a load than with too light a load, as far as the consumption of steam is concerned.

If an engine is overloaded, the surest way of improving the operation is to add a condenser, the gain of which is from 20 to 25 per cent., when account is taken of the steam-consumption of the engine only; but if measured from the coal-consumption the gain will be less, because it is not possible to heat the feed-water to so high a temperature by means of exhaust-steam when a condenser is used as when running non-condensing.

STEAM-WASTE FROM LEAKAGE

The steam-leakage past the valves and pistons of both high- and low-speed engines is of notable amount; at high speed the increase of leakage, with the greater difference of pressure on each side of the piston, is less than at low speed, and with jacketed, less than with non-jacketed cylinders. It has also been noted that good lubrication of valves and cylinders reduces the leakage materially.

In experiments made to determine what effect superheating would have on leakage loss, it was found, as has been the case in some other similar experiments, that superheating would reduce the leakage loss about 25 per cent., the reason being, apparently, that a less weight of superheated steam than of saturated steam flows through a narrow fissure, and the condensation is reduced.

In trials with the valve stationary there was less leakage than when it was moving; but when the valve was moving, the leakage became less as the speed of running became greater. Experiments made with the valve stationary in different positions seem to show that the leakage is approximately in inverse proportion to the amount of overlapping of the port and valve; that is, the greater the amount of overlapping the less the leakage.

Experiments on the leakage of steam past the piston, by admitting steam to one end of the cylinder and blocking the port at the other end, and by weighing the condensation in the dead end, showed that this leak is less than 2 per cent. of the steam-consumption of the engine.

It is evident that with a valve-leakage error of from 4 to 20 per cent. and a piston-leakage error of from 1 to 2 per cent., experiments on initial condensation which do not take these factors into account will be misleading.

An unlooked-for result was the discovery that the loss due to condensation on the cylinder-walls of unjacketed engines diminishes with a rise of initial pressure and temperature, the ratio of expansion being constant, and that this law holds without regard to the speed of the engine.

THEORETICAL EFFICIENCY OF THE STEAM-ENGINE

An engine receiving all its heat at some given temperature, and rejecting the heat (not lost by expansion) at some lower temperature, must, with its conveyer, pass through a series of changes in pressure and volume, according to Carnot's cycle, without loss or gain of heat from outside sources. Such an engine would be reversible. No such engine can be constructed or practically operated; but its theoretical efficiency serves as a standard of comparison, toward which the modern ideal design and construction are tending. The efficiency of the perfect elementary engine depends only upon the highest and lowest temperatures between which it is worked, and is independent of the nature of the working substance.

The following table has been computed from the formula $\frac{T-T_1}{T}$, in which T represents the absolute temperatures of the initial and exhaust steam derived from their absolute pressures. The upper horizontal line contain the barometric negative pressures due to the absolute pressure in pounds in the second horizontal line.

A study of the above table will show approximately the saving that may be effected by reducing the back pressure of any engine, which in many cases is ignored because the effect is not readily seen. It has been observed, in trials, that the back pressure may be as great as 3 or more pounds above atmospheric pressure from the use of long or small exhaust-pipes, many elbows, defective valve-movement, or small ports or steam-passages in the cylinder.

TABLE XXIX.—THEORETICAL HEAT-EFFICIENCY OF A PERFECT STEAM-ENGINE AT VARIOUS ABSOLUTE INITIAL AND BACK PRESSURES, SHOWING PERCENTAGE OF EFFICIENCY.

Initial pressure, absolute.	27.88	25.85	23.83	21.78	19.74	17.70	13.63	9.56	5.49	0	+3.3 lbs.	+5.3 lbs.
	1	2	3	4	5	6	8	10	12	14.7	18	20
60	25.3	22.1	20.1	18.5	17.3	16.3	14.6	13.2	12.0	10.7	9.3	8.6
70	26.3	23.1	21.2	19.6	18.4	17.4	15.7	14.4	13.2	11.9	10.5	9.8
80	27.2	23.4	22.1	20.6	19.4	18.4	16.7	15.4	14.2	12.9	11.6	10.9
90	28.0	24.8	22.8	21.4	20.2	19.2	17.6	16.2	15.1	13.8	12.5	11.8
100	28.6	25.6	23.6	22.2	21.0	20.0	18.4	17.1	16.0	14.7	13.4	12.7
110	29.3	26.2	24.3	22.8	21.6	20.7	19.1	17.8	16.7	15.4	14.1	13.4
120	29.8	26.8	24.9	23.4	22.3	21.3	19.7	18.4	17.4	16.1	14.8	14.1
130	30.4	27.4	25.4	24.1	23.0	22.0	20.4	19.1	18.1	16.8	15.5	14.8
140	30.9	27.8	26.0	24.6	23.5	22.5	20.9	19.6	18.6	17.3	16.1	15.4
150	31.3	28.3	26.4	25.0	23.9	22.9	21.3	20.1	19.1	17.8	16.5	15.9
175	32.4	29.2	27.6	26.2	25.1	24.1	22.6	21.4	20.3	19.1	17.8	17.2
200	33.2	30.3	28.6	27.2	26.0	25.1	23.6	22.4	21.3	20.1	18.9	18.3
225	34.0	31.2	29.4	28.0	26.9	25.9	24.5	23.3	22.2	21.1	19.9	19.2
250	34.8	32.0	30.1	28.8	27.7	26.8	25.4	24.1	23.1	22.0	20.8	20.1
275	35.4	32.6	30.8	29.5	28.4	27.5	26.1	24.9	23.8	22.7	21.5	20.9
300	35.9	33.2	31.4	30.1	29.1	28.2	26.7	25.6	24.5	23.4	22.2	21.6

For a back pressure of 3 pounds the loss in efficiency may be 1.3 per cent. at ordinary initial pressures, more at low pressures, and less at the high pressures; but when condensation and its value are considered, the saving is very much more apparent, and becomes a strong plea in favor of the use of compound condensing-engines wherever it is possible to operate them. Since surface-condensers have become so perfected and water-cooling towers available, the compound condensing-engine has become of the first consideration in the instalment of factory and electric power.

ACTUAL EFFICIENCY

The actual efficiency of any type of steam-engine has been usually derived from the number of pounds of steam used per hour, or of water fed to the boiler, divided by the horse-power. Thus, an ordinary engine, using 500 pounds of water per hour and developing 15 indicated horse-power, will consume $\frac{500}{15}=33.3$ pounds per horse-power hour. As a horse-power corresponds to the development of 33,000

foot-pounds per minute, and as 778 foot-pounds is the equivalent of one thermal unit, then $\frac{33{,}000}{778}=42.42$ units per horse-power, which may be a constant for obtaining the thermal efficiency of the engine, and $\frac{42.42}{\text{thermal units per horse-power}}$ minute = thermal efficiency.

Then, for example, with a simple engine running with an initial pressure of 75.3 by gauge, and exhausting at atmospheric pressure, the formula for the thermal units per pound will be

(23) . . $xr+q_1-q_2$, in which x = the percentage of moisture in the steam; r = the latent heat in the steam; q_1 = the units of heat in the water at the initial pressure, and q_2 = the units of heat in the water at atmospheric pressure or at exhaust-pressure.

Using the values in the formula, we have: $.98\times888.4+291.2-180.9=980.9$ thermal units per pound; then $\frac{980.9\times33.3\text{ pounds}}{60}=544.4$ thermal units per minute, and the thermal efficiency will be $\frac{42.42}{544.4}=.077$.

The best record that we have for multicompound condensing-engines is for about 200 thermal units per horse-power minute, which shows a thermal efficiency of $\frac{42.42}{200}=.212$; and with superheating there are possibilities of from 10 to 20 per cent. additional thermal efficiency in the use of steam and a saving of from 6 to 8 per cent. in coal-consumption, depending upon the method of obtaining the superheat.

COMPRESSION AND BACK PRESSURE

From a perusal of the large amount of discussion which has pervaded the technical journals of late years, the economical value and use of compression seem to be very much tangled, although its mechanical value is generally conceded by the evidence of its usefulness as shown in actual trials, in which it has been found indispensable in high-speed engines with a graduation due to the degree of speed.

Its economy of steam and power seems to be the principal field of discussion, from which the facts should decide the points at issue. As its mechanical effect upon the momentum of the moving parts of

an engine to the extent of producing its silent action at various speeds is obvious, it only needs the computed amount formulated from the experience of trials.

The principal facts shown by the indicator-card are that the clearance- and steam-passages must be filled, for each stroke of the piston, with a volume of steam equal to the total clearance, less the exhaust-pressure; which adds 3 per cent. to the mean effective pressure at $\frac{1}{4}$ cut-off and 5 per cent. clearance. Then the total value of the clearance-volume at $\frac{1}{4}$ cut-off being but 3 per cent. of the mean effective pressure and the clearance 5 per cent. of the stroke, the loss of steam due to clearance will be $\frac{5}{30} = 16$ per cent. loss, and $16 - 3$ per cent. gain in power equals 13 per cent. loss due to clearance.

On the other hand, if compression is carried up to the initial pressure of say 100 pounds, the heat generated by compression will raise the temperature of the compressed exhaust from 213° to above 600° F., or about 260° above the temperature of the initial steam, the superheat of which will be given to the cylinder-walls of the cut-off and clearance-space. The back pressure due to compression will be fully compensated by the expansion of the accumulated pressure behind the piston for the next stroke, and the clearance-volume of initial steam will be saved. This should hold as a proportion for any degree of compression.

As excessive compression is not needed for counteracting the momentum of the moving parts for smooth running, a noted builder of high-speed engines assumes that for 130 revolutions per minute, 24-inch stroke, compression should commence at 9 per cent. from the terminal of the stroke; for 160 revolutions per minute, 24- inch stroke, 12 per cent.; and for 240 revolutions per minute, 16-inch stroke, 19 per cent.

The author suggests that a more equable ratio of compression for balancing momentum would be derived from the equation:

$$(24) \quad . \quad . \quad \sqrt{\frac{\text{rev. pr. m.}}{\text{stroke in inches}}} = \text{length of compression in inches.}$$

This may not answer fully for the difference in weight of the moving parts as designed by different builders and for different pressures.

Of late years there has been much discussion in regard to the

economy of compression and also as to its mechanical value. This discussion, and the arguments advanced for and against compression, have as yet proved nothing, and experiments so far made have shown such discordant results as to cause distrust in their methods.

Experiments in Belgium and Germany have shown a marked falling off in efficiency with heavy compression, the difference amounting in one case to an increase of 50 per cent. in the steam-consumption.

For instance, the experiments of Professor Dwelshauvers-Dery, at Liège, showed, as the compression was increased from 10 up to 30 per cent., an increase of 21 per cent. in the steam-consumption, and for a further rise of 40 per cent. compression an increase of 50 per cent. in the steam used over that with no compression. On the other hand, careful experiments at Stevens Institute and at Cornell University show only a slight change in the steam-consumption accompanying increased compression.

It is difficult to believe that any such difference as that shown by the European experiments could result from so slight a cause. The only loss that can result from an increase of compression is the loss of work shown by the rounding of the heel of the diagram, which is largely offset by the decrease in the amount of fresh steam required to fill the clearance up to the initial pressure. There is some condensation of the cushion-steam, but this helps to warm up the cylinder and piston-ends and to diminish the initial condensation.

THE ECONOMY OF HIGH-PRESSURE STEAM

The economy due to high pressure has been slowly developed in practice by its gradual increase for power during the latter part of the nineteenth century; so that the general limit of 50 to 60 pounds rose to 80, 100, and even to 160 pounds for special purposes in a single cylinder; and for multiple expansions, 150 to 200 pounds, which is probably nearing its practical limit, although 250 pounds was exploited many years since in single cylinders by Perkins, in England, with practical failure, and 1,000 pounds was used in a steam-gun of Perkins's design by the author sixty years ago in New York, which proved a practical failure for that purpose.

A standard boiler is assumed to evaporate 34.5 pounds of water per hour from and at 212° F. at atmospheric pressure, to indicate a

boiler horse-power. This rate of evaporation is approximately used for the relative size of a boiler for the required consumption of steam per horse-power in any engine. The actual evaporation at higher pressures is less by a small percentage, for its rating for at 75 pounds pressure it is 33.85 pounds, and at 150 pounds it is 32.89 pounds.

The weight of steam per cubic foot increases in a far greater ratio in a rising pressure than is due to the decrease in boiler-evaporation, being .0380 pound at atmospheric pressure, .208 pound at 75 pounds gauge, and .367 pound per cubic foot at 150 pounds gauge-pressure; inversely, the relative volumes vary greatly from 1,646 cubic feet per pound at atmospheric pressure—which is valueless as a power from pressure alone—to 299 cubic feet per pound at 75 pounds pressure, and 169 cubic feet per pound at 150 pounds pressure.

With any given single-cylinder engine using steam at 75 pounds, cutting off at $\frac{4}{10}$ with 5 per cent. clearance, the quantity of steam used per cubic foot of cylinder-volume will be $.208 \times .4 = .0832$ pound per cubic foot, with a mean effective pressure of 58.6 pounds and terminal of 25 pounds absolute. For the same cylinder using steam at 150 pounds, cutting off at $\frac{1}{10}$ with 5 per cent. clearance, the quantity of steam used per cubic foot of cylinder-volume will be $.367 \times .1 = .0367$ pound, with a mean effective pressure of 58.8 pounds and terminal of 8.4 pounds absolute, thus obtaining a saving in steam of 56 per cent. for the same power. The saving in boiler-capacity and fuel will approximate this proportion. In ordinary practice these figures may not be reached, but a great saving has been proved under practical conditions.

The losses and gains in economy of the use of steam are well illustrated by the following diagrams. In Fig. 142 is shown the loss by decrease in the ratio of expansion.

The solid outline represents the work-area due to expansion of steam when the cut-off occurs at half-stroke. That is, $gm = 2ga$, or the number of expansions is two. If, now, the number of expansions is increased to three, so that $gn = 3ga$, then there is added an area $bcdf$, shown in dotted outline, which represents an extra amount of work obtained without increasing the quantity of steam used, since there is no alteration of the volume of steam previous to cut-off.

For example, at 100 pounds absolute initial pressure, with 50 per

cent. cut-off, the mean effective pressure will be 84.6 pounds for two expansions; if expanded three times, the mean effective pressure will be

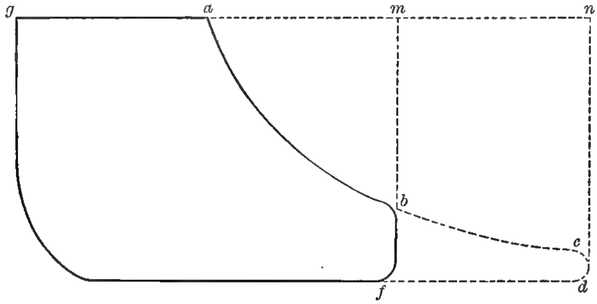


FIG. 142.—Loss in expansion.

69.9 pounds, with 50 per cent. more work at the reduced mean effective pressure, or as 84.6 is to 69.9+34.9; then $\frac{84.6}{104.8}=.807$, or nearly 20 per cent. more work for the same volume of steam.

Manifestly, then, the increase of ratio of expansion has made a greater amount of work available from the amount of steam used,

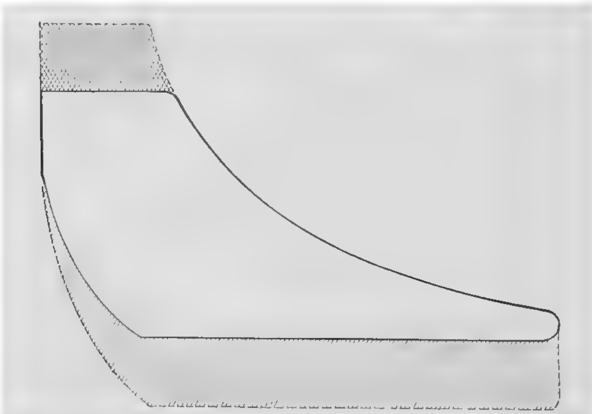


FIG. 143.—Higher pressure and the vacuum.

and it is evident that the greater the ratio the larger becomes the additional area *bcdf*, and consequently the less the steam-consumption per unit of power developed.

The ratio of expansion may be increased by increasing the initial pressure and shortening the cut-off. The final volume will thus

remain the same, but the initial volume will be less than before, and consequently the number of expansions will be greater.

Fig. 143 represents an indicator-diagram from one end of a simple non-condensing engine. Suppose that the pressure is increased 12 pounds, and cut-off shortened so that exhaust will occur at the same pressure as before. Then the steam-line will be raised to the shaded position and an extra amount of work will be obtained, represented by the area shaded with double cross-section lines. If, on the contrary, the initial pressure be left unchanged and a condenser added, so that the back pressure is reduced 12 pounds, then the area representing

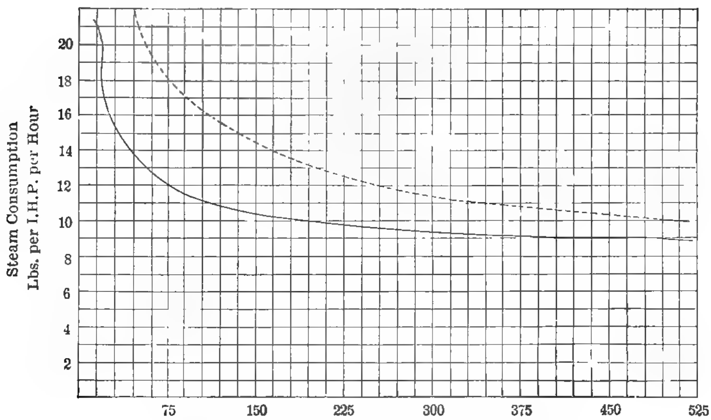


FIG. 144.—Ideal and actual curves.

increased work will be that shown by plain cross-sectioning. A comparison of the two areas, which are to the same scales, makes plain the gain to be derived from reducing the back pressure in a non-condensing engine, rather than by increasing the initial pressure.

To illustrate further the economy of the use of steam at high pressure, the diagram (Fig. 144) shows the relation of the ideal and actual curves with the steam-consumption at varying pressures in condensing-engines.

The curve shown solid is for the ideal engine, and consequently is practically valueless, inasmuch as it does not pertain to actual results obtained. But it does show what the perfect engine might accomplish, and it thus forms a basis of comparison for results

which have been secured in practice. The curve shown dotted is plotted from the results of tests made upon actual engines at various steam-pressures, the results showing highest economy being taken in plotting the curve. As can be seen, the actual curve approaches the ideal as the pressure rises, indicating that as the pressure is increased the economy of the actual engine approaches more nearly that of the ideal engine.

THE MOST ECONOMICAL POINT OF CUT-OFF

This was a much-discussed question a few years since, and the cut-off was claimed to be equal to the

$$\frac{\text{absolute back pressure}}{\text{absolute initial pressure}};$$

but as nothing was proposed in regard to the effect of the clearance in this formula it should be added to give the real cut-off. For ex-

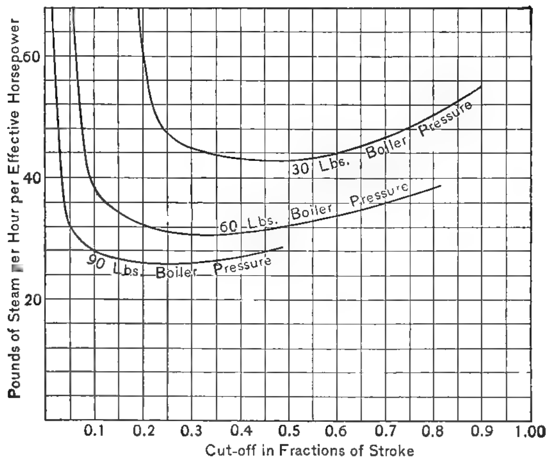


FIG. 145.—Diagram of economical cut-off.

ample, an initial pressure of 90 pounds and terminal pressure of 1 pound by gauge; this would be $\frac{15.7}{104.7} = .15$ cut-off, which would give a mean engine-pressure of 44 pounds without clearance, and 52 pounds with 10 per cent. clearance, with a corresponding increase of steam equal to a 20-per-cent. cut-off without clearance.

The experiments of Professor Denton show a larger cut-off for the above conditions by a possible addition of the clearance to the theoretical cut-off as above given.

In Fig. 145 is a diagram of the curves showing the most economical cut-off at different pressures and the consumption of steam corresponding with the cut-off. The engine was a 17×30-inch non-condensing, double-valve type, with clearance stated to be large.

In the diagram the vertical lines represent the cut-off, and the horizontal lines the pounds of steam consumed per effective horse-power. The intersection of the curves with the vertical lines shows the variation of the weight of steam for each advance in the point of cut-off. It will be seen from tracing the curves that the best result for 30 pounds pressure was obtained at about $\frac{45}{100}$ cut-off, that for 60 pounds at about $\frac{33}{100}$ cut-off, and that for 90 pounds at about $\frac{25}{100}$ cut-off. For a short distance each side of these points of cut-off the economy shows but little variation, and that with increasing pressure the point of economical cut-off has an inverse decreasing ratio.

From these and other experiments a formula has been deduced for approximately the most economical cut-off for a non-condensing simple engine, in which the cut-off $=\dfrac{100}{42\sqrt{P}}$, in which P is the initial gauge-pressure, or above the atmospheric, and above a vacuum for a condensing-engine.

The cut-off in a single-cylinder engine is limited, by an initial gauge-pressure of about 110 pounds with 5 per cent. clearance, to a minimum of one-fifth of the stroke for economic effect, as in this case the terminal pressure will be but 1 pound above atmospheric and will about equalize engine-friction.

CHAPTER XIII

THE INDICATOR AND ITS WORK

THE means of knowing what are the steam conditions within the cylinder of an engine is a most important one to all concerned in the operation of steam-power.

The interpreter is found in the indicator, a recorder of the varying pressures within the cylinder from which the action of the valves and valve-gear is noted upon sight, and by means of which the value of the steam used is made a matter of rapid computation.

The indicators in use are of several patterns, all made on the same general principle, namely, a light-moving piston, pressed by the steam against a delicate and accurately gauged spring, operating a light parallel-motion device, which marks the lines of pressure by a pencil on a paper moved too and fro, and which is placed upon a cylinder and actuated by the motion of the piston within the engine-cylinder.

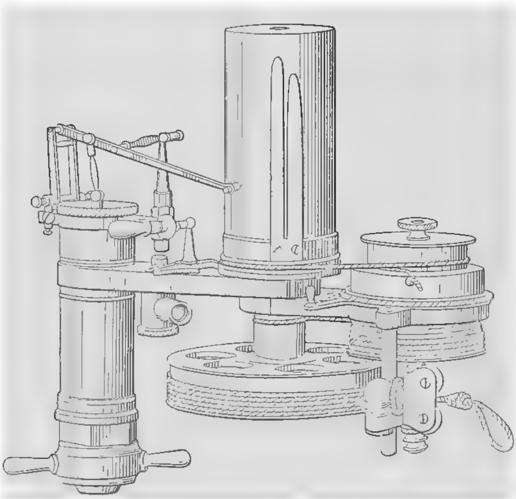


FIG. 146.—Indicator and reducing-wheel.

In Fig. 146 we illustrate one of the latest patterns of an indicator having a light aluminum reducing-wheel attached directly to the diagram-drum. The reducing-wheel has a number of bushings for its upper section, of sizes to equalize the length of diagram or card to any length of piston-stroke. This method of reducing the piston-

stroke of the engine is so neat, complete, and accurate that we forego illustration of the many awkward reduction-devices in use.

In Fig. 147 are shown the details of the construction of the Lippincott indicator. It will be noted that the indicator-cylinder is steam-jacketed, with steam-inlets below the piston so that the cylinder and piston temperatures are always the same and are also under the same pressure—a great advantage when indicating under high pressures.

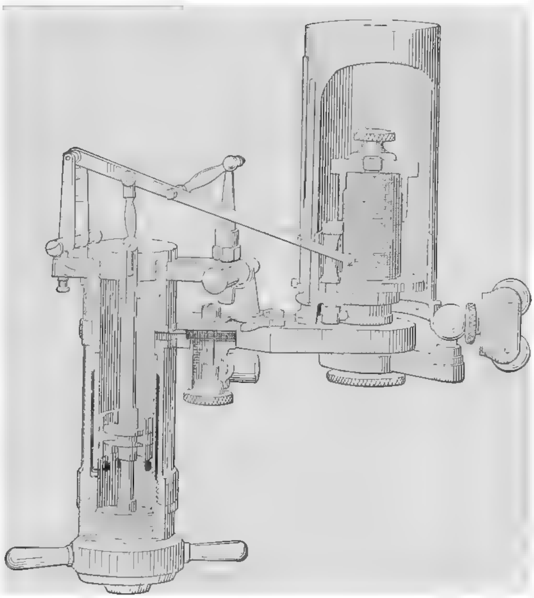


Fig. 147.—Lippincott indicator.

A special feature in its construction is the free-moving piston, which has a guide-rod to which it is fixed, with bearings at the top of the spring and in the cylinder beneath the piston, giving a perfect and free lineal motion to the piston and being practically frictionless. The piston-area is usually exactly $\frac{1}{2}$ inch for use up to 100 pounds pressure, with springs of 60, 50, and 40 pounds per inch of height in the card, while the high-pressure piston-area is exactly $\frac{1}{4}$ inch, and is suitable for indicating up to 200 pounds pressure with the No. 60 spring.

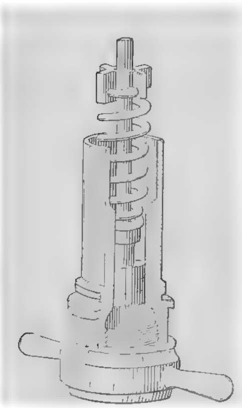


Fig. 148.—High-pressure piston.

THE ALUMINUM REDUCING-WHEEL

It is unnecessary to go into the relative merits of the reducing-wheel versus the pendulum, lazy-tongs, pantagraph, etc., as the superiority

of a good wheel over these antiquated devices is conceded by all up-to-date engineers. We admit that some reducing-wheels give so much trouble from disarrangement of cords, breakage of springs, and excessive wear that some engineers have gone back to the old methods; but we have yet to learn of a case where a user of the aluminum wheel has discarded it. By the aid of this wheel directly connected to the indicator it is possible to indicate several different engines in a day; but we have known of hours being consumed in securing material for, and rigging up, a pendulum, and often with inaccurate results.

The general design of the wheel is shown in Fig. 146. It is compact, and at the same time has not been made so small that it is subjected to rapid wear by use, or so that its application to long-stroke engines will seriously tax its capacity.

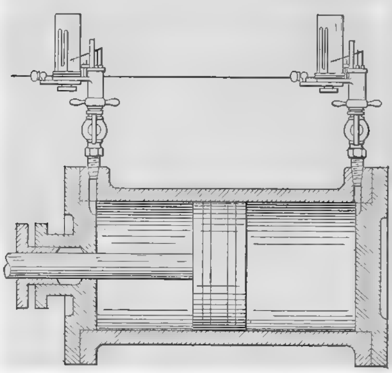


Fig. 149.—Close-connected indicators.

The main-cord wheel is made of aluminum, turned inside and out, and perfectly balanced. This wheel is capable of operating on strokes as high as $7\frac{1}{2}$ feet, or more, but by substituting a smaller pulley it becomes suitable for short strokes and high speeds, its range then being from 6 to 24 inches.

The spring-case spindle extends through the case, and is provided with a coarse square thread, eight to the inch, upon which is a suitably shaped composition-nut, to which is attached the guide-pulley arm.

Each revolution of the main-cord wheel moves the guide-pulley across the face of the wheel about $\frac{1}{16}$ inch, so that the main cord is guided perfectly on the wheel, no matter in what direction it is led.

The setting of an indicator is an important matter when accurate results are sought. The point most desired is to have a quick transit of the pressure in the cylinder to the piston of the indicator, and for this purpose the indicator should be attached directly to the clearance-space, of the cylinder without piping, elbows, or cocks,

save the one on the indicator. This is most desirable on high-speed engines, as shown in Fig. 149; but as this needs two indicators, the usual way for a single indicator is to connect the clearance-spaces with a cross-pipe, with the indicator in the centre, as shown in Fig. 150. When the connecting-pipes are desired to be retained, the angle-cocks at each end are needed to lessen the clearance. In this case, with long-stroke engines the cross-pipe should be one size larger than the thread of the indicator-cock; but a better way is to use two indicators.

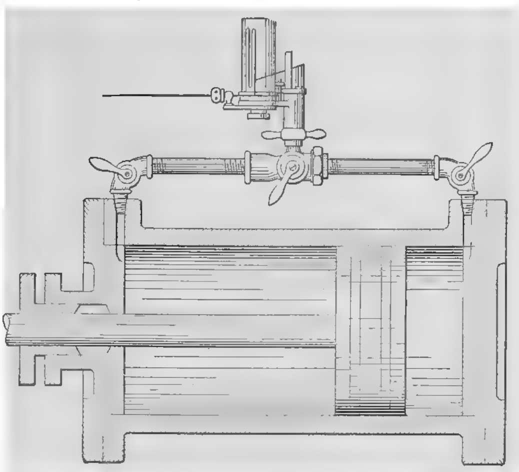


Fig. 150.—Cross-pipe connection.

A too long pipe-connection or a too small one produces freak cards, which do not represent the true action of the steam within the cylinder. On Corliss and other four-valve engines the indicator is attached at the side of the cylinder, as shown in Fig. 151. In connections to vertical cylinders care should be had to prevent water from filling the indicator-connection, as it tends to produce a freak card for the lower end of the cylinder.

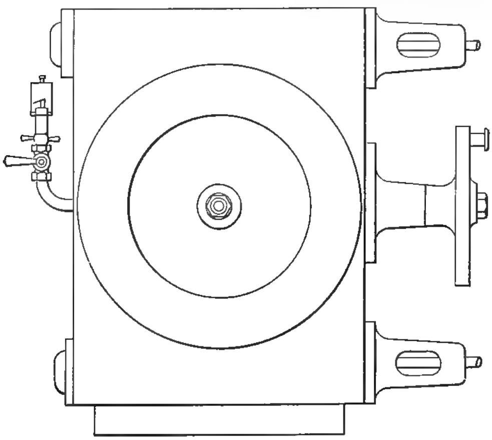


Fig. 151.—Side connection.

When a pendulum or pantagraph reducing-gear is employed a light and very flexible spring may be used to advantage to keep the cord uniformly taut. An arrangement of this kind is shown in Fig. 152, in which a loop or hook

attached to the cord may be extended by a light cord to a spring at the side of the cylinder or engine-frame—a needed arrangement for high-speed engines.

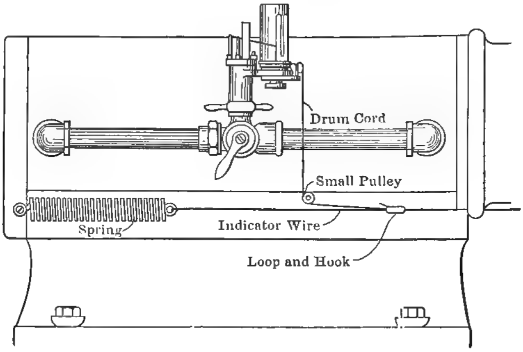


FIG. 152.—Slack-spring attachment.

Indicators are made right-and-left-hand, or with adjusting parts to make the same instrument set both ways, as will be seen in Fig. 153, in which the pencil-holder may take the pencil in opposite directions, the stop-screw being changed from one to the other side of the arm, and the drum shifted to notches provided for the change.

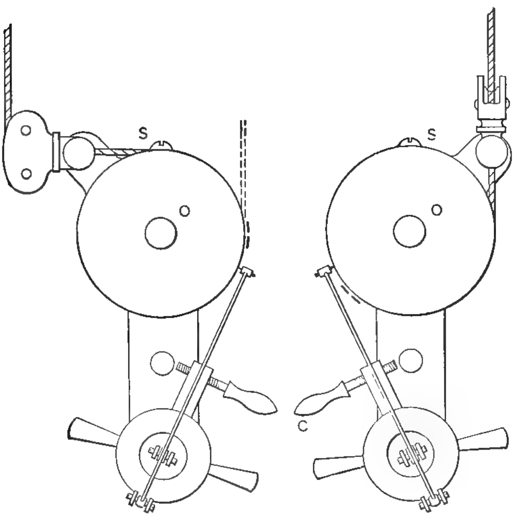


FIG. 153.—Right- and left-hand indicator.

The details for operating the various makes of indicators are sent with the indicators, and much of their mechanism becomes apparent to engineers on inspection. The pencil should be hard and sharp and the paper hard, cold-pressed letter-sheet or bond-paper, which gives a good marking with the lightest pressure of the pencil-arm.

MEASUREMENT OF THE INDICATOR-CARD

The method of measuring the mean engine-pressure from the indicator-diagram is shown in the double card (Fig. 154), which is the usual way for taking the card for both forward and back strokes. To lay off the diagram for measurement, run off on the straight edge of a piece of paper with a dividers, eleven spaces that will overrun the length of the diagram. Draw vertical lines at both ends of the diagram and two lines below it, parallel with the atmospheric line, for a register of the measurement. Lay the scale diagonally across the diagram, as shown in the illustration, at an angle that will just divide the end-spaces over the vertical lines at each end of the diagram; then mark

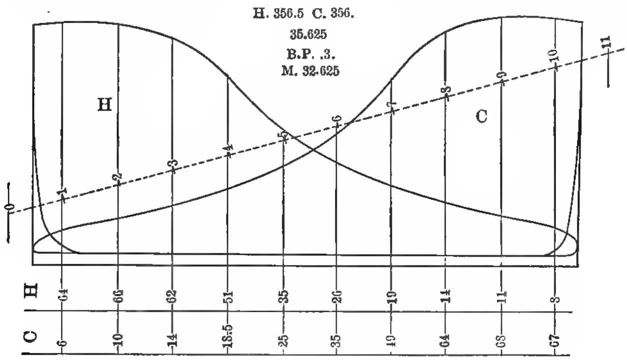


FIG. 154.—Diagram lay-out.

with a point or pencil on the diagram the ten divisions on the scale, and draw vertical lines across the marks, continuing them over the outside register-spaces. Then proceed to measure, with the scale corresponding with the indicator-spring, between the steam- and expansion-lines and the exhaust line and compression line. Enter the amounts under the heads H and C in the columns below. Divide their sums by 10 for the mean forward pressure of head- and crank-ends, and equalize for their combined mean forward pressure, less the back pressure; from which the horse-power may be computed, and from the established point of cut-off the steam-consumption may be found, as described in previous sections of this work. See "Compression" and "Steam Used per Horse-Power."

THE PLANIMETER AND THE MEASUREMENT OF THE INDICATOR-DIAGRAM

The most perfect measurement of the area and mean pressure of an indicator-diagram may now be made by the use of the planimeter which has been perfected in all its details.

In Fig. 155 is shown the Amsler planimeter, which consists of two legs jointed with points at their ends, one of which is fixed, and the other, the tracer, is moved over the diagram in the same direction as the indicator-pencil. At their juncture is a small shaft with a sharp-edged disk, a cylindrical section with a graduated scale read from a fixed vernier scale. A worm-screw and index-wheel indicate the number of revolutions of the rolling disk. To operate this planimeter, set the stationary point at any position, so that the tracing-point can be carried around the line of the diagram without bringing the wheel in contact with the paper on which the diagram is traced—preferably so that the leg with the tracer in moving around the diagram will cover an angular space between 30 and 90 degrees from the stationary pointer-leg.

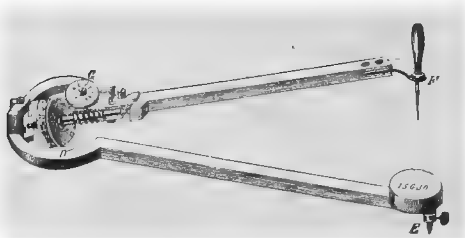


FIG. 155.—Amsler planimeter.

For the mean effective pressure divide the area as indicated by the scale by the length of the diagram in inches, and multiply the quotient by the scale of the spring used in the indicator.

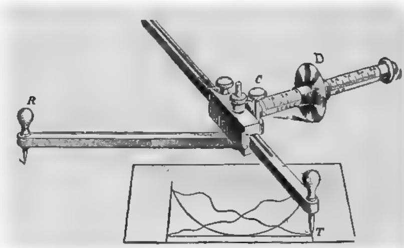


FIG. 156.—Lippincott planimeter.

One of the models of the Lippincott planimeter is shown in Fig. 156, in which *R* is the stationary point; *T* the tracer; *c* a smooth, round arm on which a scale is laid off; *D* a disk with a free motion on the scaled arm. The traverse of the wheel on the scale indicates the area.

In Fig. 157 is shown the Lippincott simplex planimeter in position

for tracing the area of an indicator-diagram. This model eliminates any possible error due to the looseness of the traversing-wheel in Fig. 156, inasmuch as the wheel is fixed on a small shaft which travels under roller-bearings at either end of the frame; so that the plane of the wheel is rigidly at right angles to its axis and therefore registers without error.

To use the simplex planimeter a large sheet of smooth cardboard should be obtained and the instrument placed on the diagram, about as shown in the figure, with the tracer-point B at either of the points

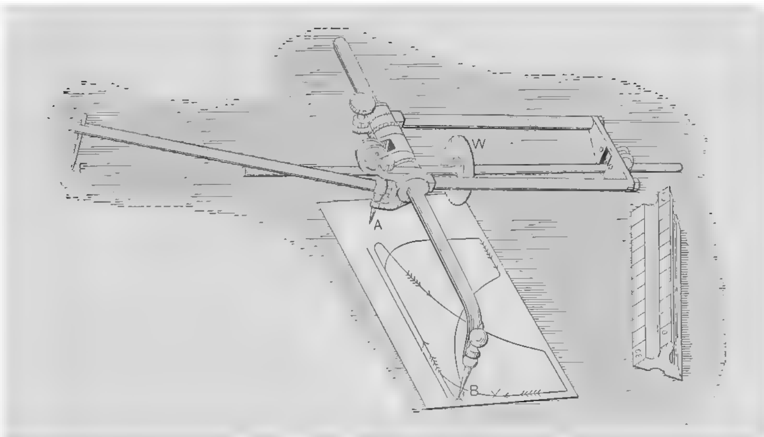


Fig. 157.—Simplex planimeter.

X, and the wheel W about $\frac{3}{4}$ inch from the body of the instrument (this distance is not important, only that the wheel must not strike the frame at either extreme of its travel). The position of the pivot-point F should be particularly noticed, the angle of the arm being somewhat greater than a right angle.

While the planimeter is held in the position shown in the figure, a slight pressure on the wheel W makes an indentation in the paper which is easily seen. The diagram is then traced in the direction of the arrows, until the tracing-point returns to the starting-point X, and while in this position the wheel W is again pressed in the paper, thereby leaving two indentations. The distance between these, measured by a scale of the same dimensions as that of the indicator-spring—a 60 scale for a 60 spring—gives the mean effective pressure direct and accurately. For the mean effective pressure direct, the

tracer-bar should be extended until the points A and B are the same distance apart as the extreme length of the diagram. If the reading is desired in square inches and tenths, the points A and B should be set 6 inches apart and a 60 scale used for reading the area in square inches and tenths.

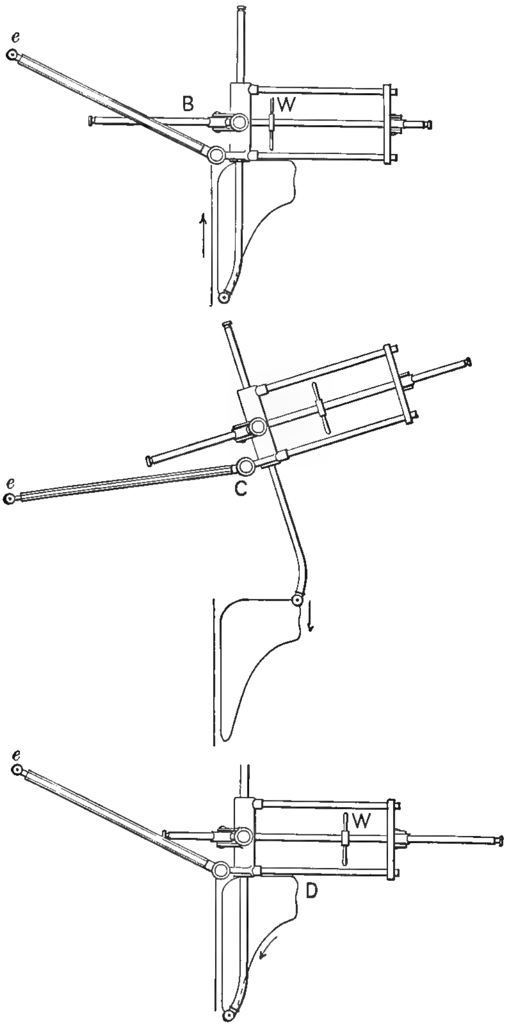


FIG. 158.—Positions of the planimeter.

The three positions of the planimeter, shown in Fig. 158, are those assumed during the tracing of the diagram.

No attention need be given to the movement of the wheel while the card is being traced, except that the wheel is clear from the frame by $\frac{3}{4}$ inch at the start, B, and does not run against the other end of the frame at the finish, D, by an extra large diagram. As the wheel rolls—when the motion is parallel to the tracer-bar—and as the shaft slides under the roller-bearings without friction for all movements at right angles to the tracer-bar, there is no scraping on the paper, so that the line may be traced without varying resistance, and the personal error due to the operator is materially reduced.

WATER USED PER HORSE-POWER HOUR

The indicator-card, besides giving the horse-power at which the engine was working when the card was taken, and the adjustments of valves, also indicates how much water the engine is using per hour. This amount is usually then reduced to the amount of water required by the engine per horse-power per hour, so that it can be compared to other engines, as each engine runs under varying conditions and may be of a different type. This unit is taken as a standard of comparison.

The indicator-card assumes that all the steam within the cylinder is steam, and takes no account of initial condensation or condensation during expansion. It also does not take into account any leakage, either through the valves or past the piston. The only way that the actual amount of water that an engine uses can be obtained is by direct measurement. This is done either by condensing the steam after it has passed through the cylinder, and weighing it, or by weighing the water before it enters the boilers on its way to the engine. However, indicator-cards must be taken, from which the horse-power is obtained, and as the amount of water that the engine uses in one hour can be measured, by dividing that quantity by the horse-power the real amount of water that the engine uses per horse-power per hour can be obtained. This is usually a very laborious and painstaking process, and the necessary appliances and apparatus are rarely at the disposal of the engineer.

It is for this reason that the indicator-cards are used for this purpose, giving as they do an indication of the amount of water used by the engine, and therefore the economy. An engine always uses more water than that represented by the indicator-diagram, but never uses less water than that shown by the diagram, so that if the diagram shows an uneconomical consumption, the engine is sure to be uneconomical; but if the indicator-card shows that it is economical, it may or may not be true. This latter condition depends upon whether there is much initial condensation or much leakage. The more leakage and initial condensation, the more will the theoretical amount differ from the actual amount. There are always some leakage and initial condensation present in every engine, and it is for this reason that an indicator-card represents the least amount of water that the engine can use.

The method for finding this amount is explained as follows: The clearance-space of the engine should be known. By clearance in this case is not meant the mechanical clearance between the head of the cylinder and the piston when the latter is at the end of the stroke, but the volume of steam that is required to fill the valve-passages plus this mechanical clearance.

The data that should be known about the card are its length, the scale of the indicator-spring with which it was drawn, and the horse-power, which can be obtained from the area of the card by the usual formula:

$$\text{Horse-power} = \frac{\text{PLAN}}{33{,}000},$$

where P = mean effective pressure in square inches; A = area of cylinder in square inches; L = length of stroke in feet; N = number of strokes per minute.

The mean effective pressure is obtained by multiplying the area of the card by the scale of the spring and dividing the product by the length of the card;

$$\text{or, } P = \frac{\text{area of the card}}{\text{length of the card}} \times \text{scale of the spring.}$$

The most accurate method of procedure is to assume some point on the expansion-line of the card, as at A (Fig. 159), and find the pressure that corresponds to it. The line A*m* represents the position of the piston at the point A. As no more steam can enter the cylinder from the boiler after cut-off, any point can be taken on the expansion-curve after cut-off. The percentage of clearance being known, the line OG is erected at a distance from the admission-line equal to that percentage of the length of the card. That is, if the clearance is $3\frac{1}{2}$ per cent., the distance C is $3\frac{1}{2}$ per cent. of the distance L. Next find the pressure of the steam at point A. This is obtained by measuring the height A*m*, and multiplying it by the scale of the spring.

Assuming for the diagram an initial gauge-pressure of 145 pounds, card = 4.08 inches in length, exhausting on or near the atmospheric line; $x = .75$ inch, and $\frac{.75}{4.08} = .183$ cut-off. Gauge-pressure at A = 120 pounds; gauge-pressure at B, 60 pounds; and the mean effective pressure is found, as before described, to be 41 pounds per square inch. Then

for 1 cubic foot of steam in the cylinder, the weight at 120 pounds gauge by the steam-table is .304 pound. As the proportion of a cubic foot contained in the rectangle $x\text{A}m$ is $\frac{.75}{4.08} = .183$ per cent. of a cubic foot, then $.304 \times .183 = .05563$ pound, and $\frac{.05563 \times 60 \times 33{,}000}{41 \times 144} = 18.656$ pounds per horse-power hour. Also: $\frac{60 \times 33{,}000}{144} = 13{,}750$, a constant, and $\frac{.05563 \times 13{,}750}{41} = 18.656$ pounds per horse-power hour, as before.

Then for any size cylinder under these conditions, its volume in cubic feet multiplied by the speed of the piston in feet per minute—by 60 and by .05563—will equal the total weight of steam consumed per hour.

For the additional steam required to fill the clearance between the volume due to compression and that due to the initial pressure,

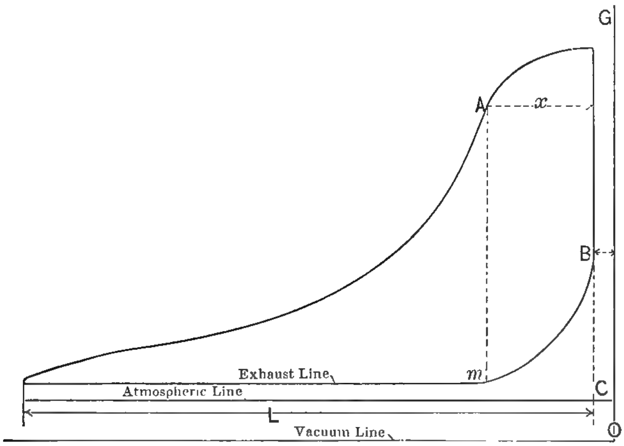


FIG. 159.—The trial indicator-diagram.

we find for 145 pounds initial pressure .356 pound per cubic foot, and for 60 pounds, .175. Then $.356 - .175 = .181 \times .035$ (clearance) $=$ $.0063 + 18.656 = 18.662$ pounds, the total weight of steam required per horse-power hour. With compressions running nearly up to the initial pressure, the differential loss is of inconsiderable value; but with no compression the loss would be $.356 \times .035 = .01246$, or, say, $1\frac{1}{4}$ per cent.

With considerable compression, and with the exhaust-line above the atmospheric line, as shown in the diagram, the computed compression may be largely increased and carried up to the initial pressure; and, inversely, when the exhaust-line is below the atmospheric line, the assumed compression is lessened.

An indicator-card from an automatic slide-valve engine, with cut-off .115, and exhausting at $2\frac{1}{2}$ pounds back pressure, is shown in Fig. 160. This engine had a 12 by 18 inch cylinder; revolutions, 124; boiler-pressure, 68 pounds; initial engine pressure, 65 pounds; com-

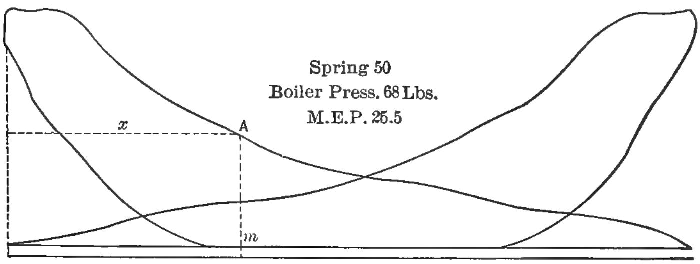


FIG. 160.—High-compression card.

pression, 57 pounds; estimated mean effective pressure, 25.5 pounds; computed horse-power, 32.5. Neglecting the clearance, which is nearly compensated by the excessive compression, and including the loss by the steam wasted in the back pressure, we find the pressure at A, from the atmospheric line, 32 pounds, at which pressure 1 cubic foot of steam weighs .113 pound.

Then $x=\frac{.112}{3.62}=.3093\times.113=.03495$ pound, the weight of steam used per cubic foot of cylinder-volume, and, using the formula, $\frac{\text{W } 13{,}750}{\text{M. E. P.}}=$ pounds per horse-power hour.

Hence $\frac{.03495\times 13{,}750}{25.5}=18.8$ pounds of steam per horse-power hour.

Then for the total steam-consumption, $18.8\times 32.5=611$ pounds per hour. With a loss of 15 per cent. by condensation during admission, then $\frac{18.8}{0.85}=22.1$ pounds per horse-power hour, or a total of 718 pounds per hour.

INDICATOR-KINKS AND ADMISSION AND TERMINAL LINES

The distortions of the lines of the indicator-card are frequently made a cause of inquiry by engineers, and for their better understanding we illustrate some of these kinks, with their accounting.

Fig. 161 is a fairly good card showing a small advance of the cut-off at the head-end over that at the crank-end, which also shows its effect on the exhaust-end by the fuller curve. The cut-off of .37 at the head-end and .34 at the crank-end shows this effect. It is not the most economical power-card, as the exhaust commences at 40 pounds and would make a better showing of steam-economy at one-fourth to three-tenths cut-off; but with an automatic cut-off these are necessarily variable points. The compression is one-half the initial pressure, or 64 pounds, which should make a smooth-running engine at the speed shown on the card.

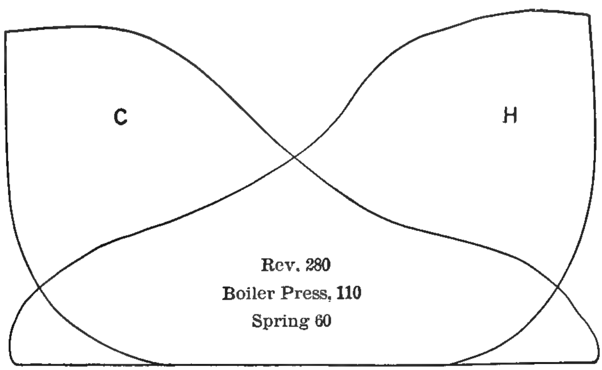


FIG. 161.—Automatic cut-off card.

The wavy expansion-lines often shown on indicator-cards are mainly due to friction in indicator parts, such as a sticky or too tight piston, looseness or tightness of the joints, too much pressure of the pencil upon the paper, rough paper, irregular tension of the barrel-spring by touching the sides of its chamber, elasticity or vibration of the cord, and the momentum of the moving parts—the last of which is greatly increased with high speed. All these produce irregu-

larities not due to valve-motion, but may sometimes become accentuated by leaky valves.

The admission- and release-lines as shown on indicator-cards,

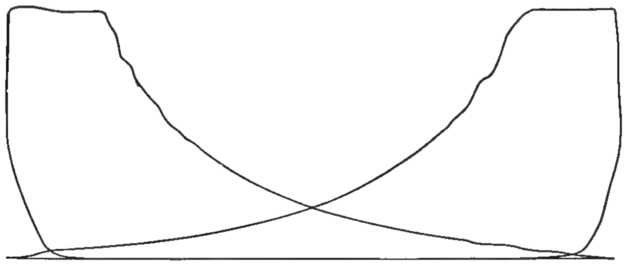


Fig. 162.—Wavy expansion-lines.

have distinct bearings on the action of the indicator, the valves, and the steam in the cylinder.

The diagram A Fig. 163 shows that at the moment when the compression-line C is completed the valve opens quickly and throws the admission-line vertically to the initial pressure. This involves the question of lead, the amount of which the size of the engine and the speed may determine. Lead should be as little as possible and allow

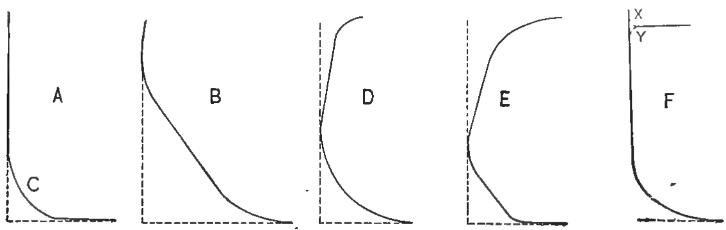


Fig. 163.—Compression- and admission-lines.

the admission to be vertical. When lead is made to admit steam just before the end of the stroke, the compression-line is carried upward, as at B. This has been a matter of discussion; but the consensus of opinion is that compression should be high with small lead, especially with high-speed engines. D and E show that the valves opened late—so much so in E as to invalidate the value of the card for economy.

The point X above the admission-line Y in the diagram F may

indicate too quick opening by lead and the momentum of the moving parts of the indicator from the sudden pressure—more often made by high-speed engines.

The steam-lines, Fig. 164, indicate variations in the normal action of valves, in which G shows a full opening to the boiler-pressure during admission, H a too small steam-pipe or excessive speed, I

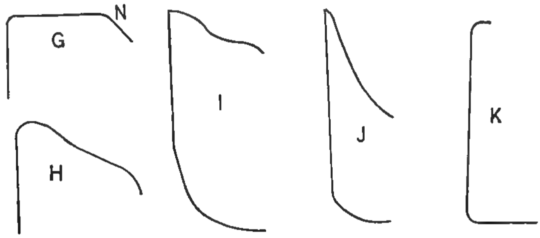


FIG. 164.—Steam-lines.

probably a large steam-chest and small steam-pipe, and J a light load and early cut-off. K shows slight compression and steam admission just past the centre—a good indication for a pounding engine.

The line L in Fig. 165 has the compression-line rising to the point C and forming a small loop, caused by late admission, the valve not opening until the return-stroke is well under way.

The diagram M shows a still later opening of the valve sometimes met with, in which the loop may vary in size and be carried to the

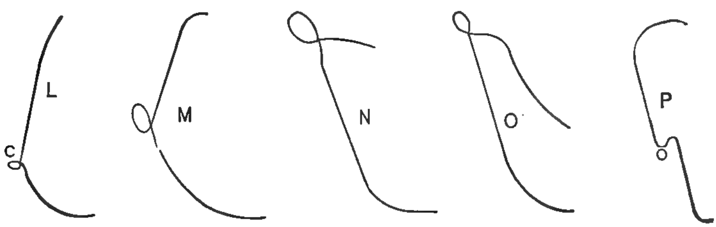


FIG. 165.—Erratic admission-lines.

top of the card, as in the diagram N, when the compression-line extends above the steam-pressure in high-speed engines from over-compression and late valve-opening. The same effect is shown in diagram O for light load and short cut-off. An offset in the compression-line,

as O at P, indicates a leak during compression by a valve lifting from its seat—generally the exhaust-valve in four-valve engines.

The diagram Q Fig. 166 shows that the exhaust closes too late to cause any compression, the piston starting on its return just before the inlet-valve opens, when steam may blow through the exhaust-valve.

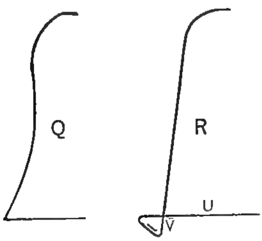


FIG. 166.—Faulty valve-setting.

In the diagram R the exhaust-valve does not close until the piston is well on its way, causing a slight vacuum before the inlet-valve opens, owing to slipping of the eccentric, thereby making the whole valve-motion late.

The forms of the release-lines are relative counterparts of the steam- and admission-lines, and are subject to abnormal proportions depending upon the steam-lines and the exhaust-valve action.

In the diagram B Fig. 167 the dropping of the expansion-line below the exhaust is very undesirable in a working engine, except in extreme conditions of load or in friction trials. The loop B varies

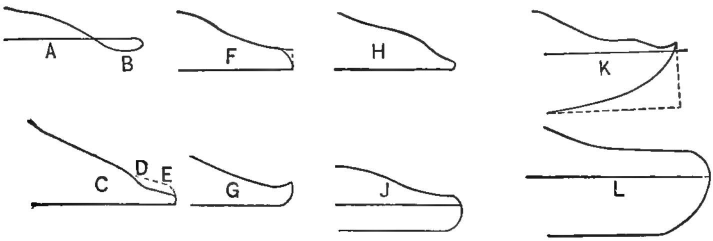


FIG. 167.—Release-lines.

in form and length by the action of the exhaust-valve. In the diagram C the release takes place at D by too early opening and throttling of the exhaust-valve, whereas the opening should be made at E, as shown by the dotted line, and a small saving made in the mean effective pressure. A better release is shown at F and H. There can be no object in delaying the release causing increase of pressure at the moment of exhaust, as shown at G and J, the latter being a late release on a condensing-engine, of which K is a good example of a slow release. The small drop near the end of the expansion-line generally shows a

slow action of the exhaust-valve, or a throttling in the exhaust-passages.

In Fig. 168 are shown two sets of cards from a high-pressure cylinder—30×48—at 87 revolutions per minute, exhausting into a

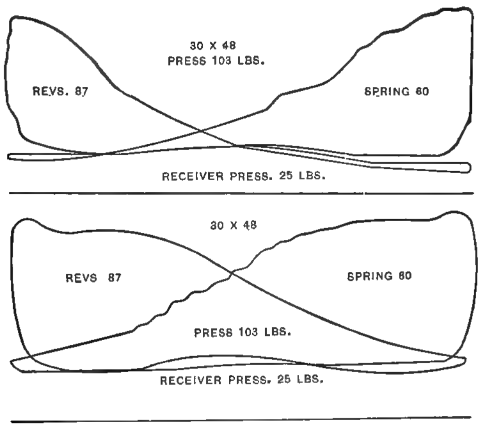


FIG. 168.—Exhaust-lines.

receiver under a varying load. The sweeping upward of the exhaust-line through the middle of the stroke shows a throttling of the exhaust-passages between the high-pressure cylinder and the receiver.

Much more might be written in regard to the eccentricities observed on indicator cards, due to the combined effects of valve-gear setting and its looseness of joints; irregularities of indicator movements and transmission devices; but we think enough has been shown to cover the leading faults, from which the origin of minor irregularities shown on cards may be readily located.

CHAPTER XIV

STEAM-ENGINE PROPORTIONS

ECONOMY in the use of steam is one of the first considerations in the design of a steam-engine. The cylinder, its principal part, should have its relative dimensions of diameter and stroke as nearly equalized as possible, unless other requirements—such as speed, or lightness of parts, or of the whole engine—suitable for its special service, may be inducements for designing the longer stroke and slower speed, as used in the comparatively slow-speed Corliss type. A large proportion of the high-speed engines of to-day are designed on the short-stroke ideal.

The considerations for the condition of the steam are essential, and may be taken as follows:

Saturated steam is steam of the temperature due to its pressure.

Superheated steam is steam heated to a temperature above that due to its pressure.

Dry steam is steam which contains no moisture, and it may be either saturated or superheated.

Wet steam is steam containing intermingled moisture, mist, or spray, with a temperature equal to that of dry saturated steam at the same pressure.

The following formulas for steam-engine proportions are quoted from various authorities, and show slight differences derived, probably, from different lines of investigation and experience in formulating values for steam-engine design. The builders of to-day may have arrived at still different values and proportions in their practice.

INITIAL CONDENSATION.

Bodmer: W = weight of steam condensed $= C \dfrac{S\,(T-t)}{L\,\sqrt[3]{N^2}}$ pounds per minute.

T = mean admission-temperature;

t = mean exhaust-temperature;

S = clearance-surface in square feet;
N = number of revolutions per second;
L = latent heat of steam at mean admission-temperature;
C = constant for any given type of engine.

For high-pressure, non-jacketed engines, C = about .11; for condensing, non-jacketed engines, C = .085 to .11; for condensing jacketed engines, C = .085 to .053. The figures for jacketed engines apply to those jacketed in the usual way, and not at the ends.

C varies for different engines of the same class, but is practically constant for any given engine.

The condensation may be, under varying conditions, from 20 to 50 per cent. of the weight of steam given to the engine.

THE CYLINDER—DIAMETER.

Bodmer: P = mean effective pressure in pounds per square inch;
L = length of stroke in feet;
A = area of piston in square inches;
N = number of strokes per minute;
r = ratio of length of stroke in inches to the distance travelled by piston in inches before cut-off;
p_1 = initial pressure in pounds per square inch;
p_2 = absolute boiler-pressure = gauge reading + 15 pounds.

If pipe from boiler is well lagged:

$$p_1 = \frac{12}{13} p_2 \text{ (Whitham)};$$

$$P = \frac{1 + \text{hyp. log. } r}{r} p_1 - \text{back pressure};$$

$$A = \frac{\text{I. H. P.} \times 33{,}000}{P\,L\,N};$$

$$\text{Diameter} = D = 2\sqrt{\frac{A}{\pi}} = 205\sqrt{\frac{\text{I. H. P.}}{P\,L\,N}}.$$

Thurston gives $D = \frac{1}{2} L$ to L.

For compound engines design the low-pressure cylinder by the same formulas used in designing a single-cylinder engine, for the total power, given initial and exhaust pressures, and total expansion. The size of the high-pressure cylinder is then determined by the table of cylinder-ratios.

Cylinder-Ratios in Compound Engines.

Grashof: $\frac{V}{v} = .85\sqrt{r}.$

Hrabak: $\frac{V}{v} = .9\sqrt{r}.$

Werner: $\frac{V}{v} = \sqrt{r}.$

Rankine: $\frac{V}{v} = \sqrt{r}.$

V = volume of low-pressure cylinder;
v = volume of high-pressure cylinder;
r = ratio of expansion.

Busley:

Boiler-pressure in pounds per square inch	60	90	105	120
$\frac{V}{v}$	3	4	4.5	5

Cylinder-Ratios in Triple-Expansion Engines.

Whitham:

Volumes, High to Low.

Boiler-pressure gauge.	High pressure.	Intermediate.	Low pressure.
130	1	2.25	5
140	1	2.4	5.85
150	1	2.55	6.9
160	1	2.7	7.25

For 170 and upward, use quadruple-expansion.

Common rule: Ratio of volumes of high to intermediate, and that of intermediate to low, are each equal to $\sqrt[3]{r}$, and the ratio of high to low $= \sqrt[3]{r^2}$.

Seaton:

Volumes, High to Low.

Boiler-pressure, absolute.	High.	Intermediate.	Low.
125	1	2	5
135	1	2	5.4
145	1	2	5.8
155	1	2	6.2
165	1	2	6.6

General practice:

Diameter of intermediate cylinder = 1.5 diameter of high;

Diameter of low-pressure cylinder = 2.5 diameter of high.

Length (as given by Whitham):

Length of bore = L + breadth of piston-ring = $\frac{1}{8}$ to $\frac{1}{2}$ inch;

Length between heads = L + thickness of piston + sum of clearances at both ends.

CYLINDER THICKNESS.

t = thickness of cylinder in inches.

Thurston: $t = a\,p_1\,D + b$:

p_1 = initial unbalanced steam-pressure in pounds per square inch;

a is a constant, equal to .0004 in short-stroke or vertical-cylinder engines, and equal to .0005 in long-stroke or horizontal-cylinder engines;

b is a constant varying from 0 to $\frac{1}{2}$ inch.

Whitham: $t = .03\sqrt{p\,D}$ for any size cylinder;

$t = .003\,D\sqrt{p}$ for small cylinders;

p = boiler-pressure in pounds per square inch.

Seaton: $t = .5 + .0004\,p\,D$.

Unwin: $t = .02\,D + .5$ to $.05\,D + .5$ (variable).

Van Buren: $t = .0001\,D\,p + .15\sqrt{D}$.

Weisbach: $t = .8 + .00033\,p\,D$.

Haswell: $t = .0004\,p\,D + \frac{1}{8}$ for vertical;

$t = .0005\,p\,D + \frac{1}{8}$ for horizontal.

Marks: $t = .00028\,p\,D$.

Rankine: $t = \frac{p\,D}{2\,f}$;

f = tensile strength of material, with a factor of safety, from 30 to 40.

Barr: $t = .05\,D + .3$ inch, a formula which represents the average practice of modern engine-builders.

CYLINDER-HEADS.

Thurston: $t = .000333\,D\,p + .25$;

D being diameter of circle in which the thickness is taken;

$t = .005\,D\sqrt{p} + .25$.

Marks: $t = .003\, D\sqrt{p}$;

$t = .00035\, p\, D$.

Seaton: $t = .0005\, p\, D + .25$;

$t = .0022\, p\, D + .93$.

Kent: $t = .00036\, D\, p + .31$, which represents average practice.

CYLINDER-HEAD BOLTS.

Whitham: Diameter of bolt-circle = D + twice the thickness of the cylinder + twice the diameter of bolts.

Bolts should not be more than 6 inches apart.

$d = D\sqrt{\frac{p}{5{,}000\, n}}$;

d = diameter of bolts at root of thread;

n = number of bolts.

Marks: $n = .0001571\, \frac{D^2 p}{c}$;

c = area of one bolt.

Thurston: Distance between bolts = four to five times thickness of flanges;

$n = .0002\, \frac{D^2 p}{d^2}$.

Barr: $n = .7\, D$;

$d = .025\, D + .5$.

Both of Barr's formulas represent average practice among builders of modern low-speed engines.

CYLINDER-FLANGES.

Thurston: Thickness of cylinder-flanges is usually made equal to the thickness of flanges of the heads.

Barr: Thickness of flanges = 1.2 times the thickness of the cylinder.

CLEARANCE.

Seaton: $\frac{1}{8}$ to $\frac{3}{8}$ inch for roughness of castings, and $\frac{1}{16}$ to $\frac{1}{8}$ inch for each working-joint.

STEAM-PIPE.

Kent: Pipe-diameter = $.408\sqrt{\text{H. P.}}$

EXHAUST-PIPE.

Area = 25 to 50 per cent. greater than area of steam-pipe.

VALVE-PORTS.

Kent: Length of port = .8 D;

Area of steam-port $= \frac{A\,L\,N}{6{,}000}$ in feet;

Area of exhaust-port = 1.5 area of steam-port.

Barr: Area of steam-port $= \frac{A\,L\,N}{c}$,

where c = 5,500 in high-speed engines,
and c = 6,800 in low-speed engines.

STEAM-CHEST.

Thurston: Thickness $= .003\ D\sqrt{p}$.
Seaton: Thickness $= .7\ (.25 + .0005\ p\ D)$.
Number and size of bolts are to be determined as for cylinder-head.

VALVE-STEM.

Whitham: Diameter $= \frac{1}{30}$ D;
$= \frac{1}{3}$ diameter of piston-rod.

PISTON.

Marks: Thickness of piston-head $= \sqrt[4]{L\,D}$.
Barr: Piston-face = .46 D for high-speed engines;
= .32 D for low-speed engines.
Whitham: Thickness of piston = breadth of ring + thickness of flange on one side to carry the ring + thickness of follower-plate.

PISTON-RINGS.

Seaton: $w = .63\ (.02\ D\sqrt{p} + 1)$.
Whitham: $w = .15\ D$.
Unwin: $w = .014\ d + .08$.
Kent: $t = .0333\ D + .125$;
w = width of ring in a direction parallel to the axis of the cylinder;
t = thickness of ring on a radial line.

ECCENTRIC PISTON-RINGS.

Maximum thickness = .05 D;
Outside diameter of ring = 1.05 D;
Inside diameter of ring = .97 D;
Eccentricity of inner circle = .01 D;
$w = \frac{3}{8}$ to $\frac{5}{8}$ inch.

Kent: Mean thickness = .0333 D + .125;
Minimum thickness = $\frac{2}{3}$ maximum.

PISTON-ROD.

Unwin: $d'' = b\,D\sqrt{p}$;
p = maximum unbalanced pressure in pounds per square inch;
b = .0167 for iron and .0144 for steel;
$d'' = k\,D\sqrt{p}$;
k is a constant, depending on the stress f, allowed in the material as follows:

f	2,000	2,500	3,000	3,500	4,000
k	.0224	.02	.0182	.0169	.0158

f = 3,000 to 3,600 in short-stroke direct-acting engines;
f = 2,000 to 2,500 in long-stroke horizontal engines.

Thurston: $d'' = \sqrt[4]{\frac{D^2 p\,L^2}{a}} + .0125\,D$;
a = 10,000 in high-speed engines and 15,000 in low-speed engines.

Marks: $d'' = .0179\,D\sqrt{p}$ for iron;
$d'' = .0105\,D\sqrt{p}$ for steel.

Seaton: $d'' = \frac{D}{F}\sqrt{p}$;
F = 45 to 60 for direct-acting engines.

Whitham: $d'' = k\,D$;
k = .1 for wrought iron on condensing-engine;
= .08 for steel on condensing-engine;
= .125 for wrought iron on non-condensing engine;
= .10 for steel on non-condensing engine.

Kent: $d'' = .013\sqrt{D\ l\ p}$; l = length in inches.

Barr: $d'' = .145\sqrt{D\ L}$ for low-speed engines; $d'' = .11\sqrt{D\ L}$ for high-speed engines. $\}$ L = inches.

CROSS-HEAD SLIDES.

The thrust on the guide when the connecting-rod is at its maximum angle with the line of the piston-rod = P, tangent of the angle Z, whose sine

$= \dfrac{\text{stroke of piston}}{2 \times \text{length of connecting-rod}}$;

$P = .7854\ D^2 p$;

Area of slide $= \dfrac{P \text{ tangent } Z}{p_0}$;

p_0 = allowable pressure per square inch on slide.

Seaton: $p_0 < 400$ pounds per square inch.

Rankine: $p_0 = 72.2$ pounds per square inch.

Whitham: $p_0 = 100$ pounds per square inch.

Thurston: $p\ V < 60{,}000$ and $> 40{,}000$;

V = relative velocity in feet per minute of the rubbing-surfaces.

Barr: Area = .63 A for high-speed engines;
= .46 A for low-speed engines.

CROSS-HEAD PIN.

Seaton: Projected area $= \dfrac{.7854\ D^2\ p}{1{,}200}$;

Small engines $\begin{cases} \text{Length} = 1.4 \text{ diameter of piston-rod.} \\ \text{Diameter} = 1.25 \text{ diameter of piston-rod.} \end{cases}$

Whitham says the bearing-surface is found by the formula for crank-pin design.

Barr: Projected area = .08 A for high-speed engines;
= .07 A for low-speed engines.

CONNECTING-ROD.

Ratio of length of connecting-rod to stroke:

Thurston: 2 or $2\frac{1}{2}$ to 1.

Whitham: 2 to $4\frac{1}{2}$.

Marks: 2 to 4;

d'' = diameter of circular connecting-rods larger at the middle.

Whitham: d'' (at middle) $= .0272 \sqrt{D\ l \sqrt{p}}$;
l = length of connecting-rod in inches;
d'' (at necks) = 1 to 1.1 diameter of piston-rod;
Diameter at the crank-pin end = 1.08 times the diameter at the cross-head end. The rod is larger at the middle and tapers about $\frac{1}{8}$ inch to the foot.

Marks: $d'' = .0179\ D \sqrt{p}$, if diameter is greater than $\frac{1}{24}$ length;
$d'' = .02758 \sqrt{D\ l \sqrt{p}}$, if the diameter found by the previous formula is less than $\frac{1}{24}$ length.

Thurston: d'' (at middle) $= a \sqrt{D\ L \sqrt{p}} + C$;
$a = .15$ and $C = .5$ for fast engines;
$a = .08$ and $C = .75$ for moderate speeds.

Donaldson: d'' (at necks) $= .0024\ D^2\ p$.

Sennett: d'' (at middle) $= .01818\ D \sqrt{p}$;
d'' (at necks) $= .01666\ D \sqrt{p}$.

Seaton: $d'' = .02758 \sqrt{D\ l \sqrt{p}}$.

Kent: $d'' = .021\ D \sqrt{p}$.

Barr: $d'' = .092 \sqrt{D\ l}$.

Rectangular Connecting-Rod.

Thurston: $t = .0209 \sqrt{D\ l \sqrt{p}} + .47$;
t = distance between parallel sides;
Depth at cross-head end = 1.5 t;
Depth at crank-end = 2.25 t.

Kent: $t = .01\ D \sqrt{p} + .6$.

Bolts in End of Connecting-Rod.

Whitham: Diameter at root of thread

$$= \frac{D}{30} \sqrt{\frac{n\ p}{\sqrt{n^2 - 1}}} \text{ for wrought iron;}$$

$$= \frac{D}{37} \sqrt{\frac{n\ p}{\sqrt{n^2 - 1}}} \text{ for steel;}$$

$$n = \frac{\text{length of connecting-rod}}{\text{length of crank}};$$

p = maximum pressure.

CAP ON END OF CONNECTING-ROD.

Whitham: Depth of cap at centre

$$=2.7\ l\ \sqrt[3]{\frac{n\ D^2\ p}{b\ E\ \sqrt{n^2-1}}}\ \text{for rigidity;}$$

$$\text{Depth of cap at centre} = 1.5\ D\ \sqrt[3]{\frac{n\ p\ l}{f\ b\ \sqrt{n^2-1}}};$$

l = length of cap between bolt-centres;
b = breadth of cap;
E = modulus of elasticity of metal used:
= 28 million for wrought iron,
42 million for steel, and
18 million for cast iron;
$b = \frac{1}{2}$ to $\frac{1}{3}$ length of journal = diameter of neck of connecting-rod + $\frac{1}{4}$ to $\frac{1}{2}$ inch;
d = depth of cap = .6 diameter of connecting-rod = .8 diameter of bolts + $\frac{\text{pitch of thread on bolt}}{10}$.

CRANK-PIN.

Marks: $l = .0000247\ f\ p^1\ N\ D^2 = 1.038\ f\ \frac{\text{I. H. P.}}{L}$.

Whitham: $l = .9075\ f\ \frac{\text{I. H. P.}}{L}$;

l = length of crank-pin journal in inches;
p^1 = mean pressure in cylinder in pounds per square inch;
f = coefficient of friction, from .03 to .05 for perfect lubrication, and from .08 to .1 for imperfect lubrication.

Thurston: $l = \frac{P\ N}{1{,}200{,}000}$ for steel pins;

$l = \frac{P\ N}{600{,}000}$ for iron pins;

$l = \frac{P\ V}{60{,}000\ d}$;

V = velocity of rubbing-surface in feet per minute;
P = mean total load on pin;
d = diameter of pin.

Rankine: $l = \dfrac{P\ (V+20)}{44{,}800\ d}$.

Unwin: $l = \dfrac{P}{p_0\ d}$;

P = greatest load on pin;

p_0 = pressure in pounds per square inch of bearing;

p_0 varies from 150 to 200 in small land engines, and from 400 to 800 in large land and marine engines;

$l = a\ \dfrac{\text{I. H. P.}}{r}$;

r = crank-radius in inches;

a = .3 to .4 for iron pins;

a = .066 to .1 for steel pins.

Unwin: $d = \sqrt[4]{\dfrac{5.1}{p_0\ f}}\ \sqrt{P}$;

f = twisting-stress, say 5,000 pounds per square inch.

Thurston: $d = \sqrt[3]{\dfrac{5.1\ P\ l}{9{,}000}}$ for wrought iron; increase d 10 per cent. in case of steel;

P = maximum load on the piston.

Unwin: $d = .0947$ to $.0827\ \sqrt[3]{P\ l}$ for wrought iron;

$d = .0827$ to $.0723\ \sqrt[3]{P\ l}$ for steel.

Marks: $d = .066\ \sqrt[4]{P\ l^3\ D^2} = .945\ \sqrt[4]{\dfrac{\text{H. P. } l^3}{L\ N}}$;

P = maximum steam-pressure in pounds per square inch.

Whitham: $d = .0827\ \sqrt[3]{P\ l} = 2.1058\ \sqrt[3]{\dfrac{2\ \text{H. P. } l}{L\ N}}$;

$d = .405\ \sqrt[4]{P\ l^3}$.

Barr: Projected area = a = .24 A for high-speed engines;

a = .09 A for low-speed engines;

$l = .3\ \dfrac{\text{H. P.}}{L} + 2.5$ for high-speed engines;

$l = .6\ \dfrac{\text{H. P.}}{L} + 2$ for low-speed engines.

CRANK.

Thurston: $b = \frac{.7854 \text{ f } D^2 \text{ p R secant z l}}{\text{a } d^2};$

b = thickness of web
d = width of web;
l = radius of crank;
p = maximum unbalanced pressure in pounds per square inch;
z = angle of rod with centre-line of the engine;
f = factor of safety.

The diameters of the hubs are about twice the diameters of the corresponding shafts, and d, at either end, is three-fourths the diameter of the adjacent hub.

Empirical rules adopted by builders give for wrought iron:

Hub-diameter = 1.75 to 1.8, the least diameter of that part of the shaft carrying full load;
Eye-diameter = 2 to 2.25 times the diameter of the inserted portion of the pin;
Hub-depth = 1.0 to 1.2 diameter of shaft;
Eye-depth = 1.25 to 1.5 diameter of pin;
Web-width = .7 to .75 width of adjacent hub or eye;
Web-depth = .5 to .6 depth of adjacent hub or eye.

Whitham: $W \text{ x} = \frac{\text{f k } (2 \text{ y})^2}{6};$

k = thickness parallel to shaft, and generally = .75 shaft-diameter;
2 y = variable width;
x = distance from crank-pin centre to place where 2 y is measured;
f = allowable stress per square inch of material;

$$W = .7854 \text{ } D^2 \text{ p } \frac{\sqrt{n^2+1}}{n},$$

$$\text{where } n = \frac{\text{length of connecting-rod}}{\text{length of crank}}.$$

For a two-cylinder engine W = 1.414, and for a three-cylinder engine W = 2 times the above value.

SHAFT.

When designed for combined twisting and bending:

Whitham: $d = \sqrt[3]{\frac{5.1\ T'}{f}}$;

T = greatest twisting-moment on shaft due to load on the piston;

M = greatest bending-moment on shaft due to load on the piston;

T' = Equivalent twisting-moment $= M + \sqrt{M^2 + T^2}$ on outer journal for overhung crank;

$T' = \frac{M}{2} \sqrt{\frac{M^2}{4} + T^2}$ for double, crank arm.

The above formula gives safe values except with very heavy fly-wheels, where the shaft must be designed with reference to bending due to the weight of fly-wheel and shaft.

Kent: $d = .43\ D$ for long-stroke engines;

$d = .4\ D$ for short-stroke engines.

For two cranks at 90 degrees:

$d = 1.932 \sqrt[3]{\frac{T}{f}}$;

T = maximum twisting-moment produced by one piston;

f = safe-working shearing-strength of material.

LENGTH OF SHAFT-BEARINGS.

Marks: $l = .0000247\ f\ p\ N\ D^2$;

f = coefficient of friction;

p = mean pressure in pounds per square inch on piston.

Unwin: $l = \frac{.4\ H.\ P.}{r}$;

r = radius of crank in inches;

$l = (.002\ N + 1)\ d$ for wrought iron;

$l = (.0025\ N + 1.25)\ d$ for steel.

Barr: $l = 2.2\ d$ for high-speed engines;

$l = 1.9\ d$ for low-speed engines.

FLY-WHEELS.

D_1 = diameter in feet $= \frac{3{,}820}{N}$.

Thurston: $D_1 = 4\ L$;

W = weight $= 1{,}000{,}000\ \frac{A\ L\ p}{N^2\ D_1^2}$ for automatic valve-

gear engines and ordinary forms of non-condensing engines with a ratio of expansion from 3 to 5;

p = mean steam-pressure in pounds per square inch;

$W = \frac{a\ A\ L}{N^2\ D_1^2}$;

a ranges from 40 to 60 million, with an average value of 48 million.

Rankine: $W = 1{,}900{,}000\ \frac{A\ L\ p}{V\ D_1^2\ R}$;

V = variation of speed per cent. of mean speed.

Stanwood: $W = 2{,}800{,}000\ \frac{D^2\ S}{D_1^2\ N^2}$;

D = diameter of cylinder;

S = stroke in inches.

FLY-WHEEL RIMS.

$$t = \frac{.7\ D_1}{N^2}.$$

FLY-WHEEL ARMS.

Torrey: $b = \frac{W\ L}{30\ d^2}$;

$W = \frac{S\ y}{n}$;

W = load in pounds acting on one arm;

L = length of arm in feet;

S = strain on belt in pounds per inch of width, taken at 56 for single and 112 for double belts;

y = width of belt in inches;

d = depth of arm at hub in inches = major axis;

b = breadth of arm at hub in inches = minor axis.

In using the formula assume depth.

Depth and breadth can be reduced by about $\frac{1}{3}$ at rim.

Unwin: $d = .6337\ \sqrt[3]{\frac{B\ D_1}{n}}$ for single belts;

$d = .798\ \sqrt[3]{\frac{B\ D_1}{n}}$ for double belts;

b = .4d;

B = breadth of rim.

MAXIMUM SPEED OF FLY-WHEELS.

80 feet per second = 4,800 feet per minute; $\text{R. P. M.} = \frac{1{,}527}{D}$.

88 feet per second = 5,280 feet per minute; $\text{R. P. M.} = \frac{1{,}680}{D}$.

100 feet per second = 6,000 feet per minute; $\text{R. P. M.} = \frac{1{,}910}{D}$.

MAXIMUM DIAMETER IN FEET OF FLY-WHEELS.

80 feet per second = 4,800 feet per minute; $D = \frac{1{,}527}{\text{R. P. M.}}$.

88 feet per second = 5,280 feet per minute; $D = \frac{1{,}680}{\text{R. P. M.}}$.

100 feet per second = 6,000 feet per minute; $D = \frac{1{,}910}{\text{R. P. M.}}$.

D = diameter of wheel in feet.

The following tables of the principal dimensions of high- and low-pressure cylinders have been compiled by Mr. L. L. Willard, a designer of Corliss engines, and although they may not conform to the practice of every builder, may be a good schedule of reference for the various sizes of cylinders for the high and low pressures of 150 and 50 pounds respectively.

TABLE XXX.—HIGH-PRESSURE CYLINDER-DIMENSIONS FOR 150 POUNDS STEAM-PRESSURE.

Size.	Counterbore.	Thickness of barrel.	Thickness of steam-chest wall.	Thickness of exhaust-chest wall.	Diameter of valves.	Valve-bearing.	Depth of cylinder-head.	Thickness of solid piston.	Thickness of bull-ring piston.	Diameter of steam-pipe.	Diameter of exhaust-pipe.	Drilling-circle.	Number of studs.	Diameter of studs.	Diameter of foundation-bolts.
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
12	$12\frac{1}{4}$	1	1	$\frac{7}{8}$	$3\frac{1}{2}$	$2\frac{1}{2}$	$5\frac{3}{4}$	4	$5\frac{1}{2}$	5	6	15	10	$\frac{7}{8}$	$1\frac{1}{8}$
14	$14\frac{1}{4}$	1	1	$\frac{7}{8}$	4	$2\frac{1}{2}$	$5\frac{3}{4}$	4	$5\frac{1}{2}$	5	6	17	10	1	$1\frac{1}{8}$
16	$16\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	1	$4\frac{1}{2}$	$2\frac{1}{2}$	$5\frac{3}{4}$	4	$5\frac{1}{2}$	6	7	19	12	1	$1\frac{1}{4}$
18	$18\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	1	5	3	6	5	$6\frac{1}{2}$	7	8	21	14	$1\frac{1}{8}$	$1\frac{1}{2}$
20	$20\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	1	5	3	6	5	$6\frac{1}{2}$	8	9	23	16	$1\frac{1}{8}$	$1\frac{1}{2}$
22	$22\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{8}$	6	$3\frac{1}{2}$	$7\frac{1}{2}$	$5\frac{1}{2}$	$7\frac{1}{2}$	9	10	$25\frac{3}{4}$	18	$1\frac{1}{8}$	$1\frac{3}{4}$
24	$24\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{8}$	6	$3\frac{1}{2}$	$8\frac{1}{2}$	$5\frac{1}{2}$	$8\frac{1}{2}$	10	12	$27\frac{1}{4}$	18	$1\frac{1}{4}$	$1\frac{3}{4}$
26	$26\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$6\frac{1}{2}$	$3\frac{1}{2}$	$8\frac{1}{2}$	$5\frac{1}{2}$	$8\frac{1}{2}$	10	12	29	20	$1\frac{1}{4}$	$1\frac{3}{4}$
28	$28\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	7	$3\frac{3}{4}$	10	7	10	11	13	$31\frac{3}{4}$	20	$1\frac{3}{8}$	$1\frac{3}{4}$
30	$30\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$7\frac{1}{2}$	4	10	7	10	12	14	34	24	$1\frac{3}{8}$	2
32	$32\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	8	4	10	7	11	12	14	36	24	$1\frac{3}{8}$	2
34	$34\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$8\frac{1}{2}$	4	$11\frac{1}{2}$	8	12	14	16	38	28	$1\frac{3}{8}$	2
36	$36\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{2}$	9	$4\frac{1}{2}$	$12\frac{1}{2}$	8	12	14	16	40	32	$1\frac{3}{8}$	2

TABLE XXXI.—LOW-PRESSURE CYLINDER-DIMENSIONS FOR 50 POUNDS STEAM-PRESSURE.

Size.	Counterbore.	Thickness of barrel.	Thickness of steam-chest wall.	Thickness of exhaust-chest wall.	Diameter of valves.	Valve-bearing.	Depth of cylinder-head.	Thickness of solid piston.	Thickness of bull-ring piston.	Diameter of steam-pipe.	Diameter of exhaust-pipe.	Drilling-circle.	Number of studs.	Diameter of studs.	Diameter of foundation-bolts.
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
20	20¼	1¼	1	⅞	5	3	6	6	6½	7	8	23	16	⅞	1½
22	22¼	1¼	1	⅞	6	3½	7½	6	6½	8	9	25¾	18	⅞	1¾
24	24¼	1⅜	1	⅞	6	3½	8½	6	8	9	10	27¼	18	1	1¾
26	26¼	1⅜	1¼	1	6½	3½	8½	6	8	9	10	29	20	1	1¾
28	28¼	1⅜	1¼	1	7	3¾	10	7	9	10	12	31¾	20	1	1¾
30	30¼	1½	1¼	1	7½	4	10	7	10	10	12	34	22	1	2
32	32¼	1½	1¼	1	8	4	10	8	12	12	14	36	22	1⅛	2
34	34¼	1½	1⅜	1¼	8½	4	10	8	12	12	14	38	24	1⅛	2
36	36¼	1⅝	1⅜	1¼	9	4½	12	8½	14	14	16	40	26	1⅛	2
38	38¼	1⅝	1⅜	1¼	9	4½	12	8½	14	14	16	42	28	1¼	2
40	40¼	1⅝	1½	1⅜	9	4½	12	8½	15	16	18	44	28	1¼	2
42	42¼	1¾	1½	1⅜	10	5	13	9	15	16	18	46	28	1¼	2
44	44¼	1¾	1½	1⅜	10	5	13	9	15	16	18	48	32	1¼	2¼
46	46¼	1¾	1½	1⅜	10	5	14	9½	16½	18	20	50	32	1¼	2¼
48	48½	1¾	1½	1⅜	11	6	14	10½	16½	18	20	52	32	1¼	2½
50	50½	1⅞	1⅝	1½	11	6	15	10½	17½	20	22	54	36	1¼	2½
52	52½	1⅞	1⅝	1½	12	7	15	10½	17½	20	22	56	36	1¼	2½
54	54½	1⅞	1⅝	1½	12	7	15	11	18	20	22	58	36	1¼	2¾
56	56½	2	1⅝	1½	12	7	16	12	18	20	22	60	36	1¼	2¾
58	58½	2	1⅝	1½	12	7	16	12	20	22	24	62	40	1¼	2¾
60	60½	2	1⅝	1½	12	7	16	12	20	22	24	64	40	1¼	2¾

The piston of a steam-engine has been a matter of much study with engine-designers in order to counteract the wear of both piston and cylinder from excessive frictional action. The early form of a solid bearing-surface drifted into the form of a solid piston with one or more plain and eccentric snap-rings with followers, which later developed into a composite piston of almost as many variations as there are builders of engines, for almost every designer has a kink of his own, which he always regards as the best. In Fig. 169 is shown a composite piston, consisting of a spider and a follower-plate that clamps a bull-ring, which is made adjustable

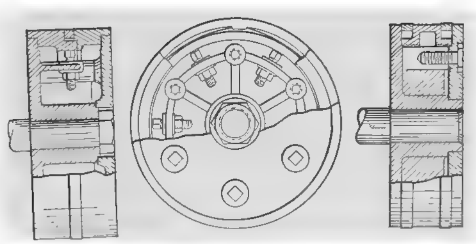


FIG. 169.—Composite piston.

by lock-screws for keeping the piston-rod concentric with the cylinder; the lock-screws are threaded in the web of the spider, with lock-nuts on the inside. On the left side of the cut is shown the single-snap spring-ring held out by flat springs against shoulders, and at the right a double spring with the same construction. This piston is used on the engines of the Murray Iron Works Company.

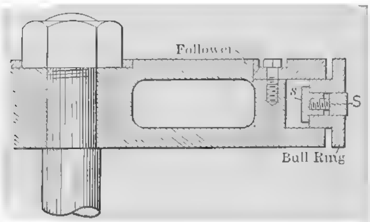


Fig. 170.—Segmental piston.

Another model, Fig. 170, has a packing-ring made in six segments, with halved joints, and set out against the cylinder-walls by small coil-springs held in brass sockets screwed into the bull-ring. The bull-ring is adjusted by lock-screws in the web of the spider.

In Fig. 171 are represented the cross-head and a half-section of the piston as made by the Hewes & Phillips Co.

The cross-head is secured to the piston-rod by a thread and nut, or a taper fit with a cross-key which draws the rod firmly to the shoulder. The wedge form of a gib at the top and bottom of the cross-head provides means for adjustment.

The sliding surfaces of the gibs are faced with antifriction metal, and the surfaces of these gibs are amply large for the severest duty.

The design of the cross-head is such as to entirely avoid the springing of the piston-rod or any tendency to force the cross-head out of line.

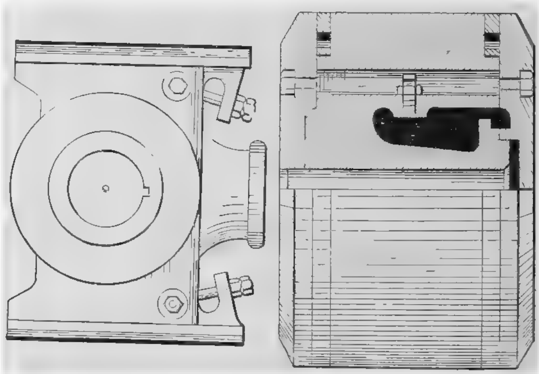


Fig. 171.—Hewes & Phillips cross-head and piston.

The piston is a strongly ribbed casting, securely fastened to the piston-rod by forcing. It is further secured by a strong and substantial key midway in its bearing in the piston. The end of the rod is also riveted. There is a tendency in all pistons to wear down or get out of centre. When this occurs the piston-rod is liable to be

grooved and the gibs of the cross-head to wear unequally on their opposite ends. To obviate this the piston is furnished with a solid bull-ring, against which suitable screws with jam-nuts are provided for adjustment. By this means perfect alignment of the piston with the cylinder can be readily secured.

The bull-ring is a solid casting turned in such a manner as to have a full semicircular bearing on the lower half of the cylinder. At either end of the bull-ring are narrow piston-rings, which wipe over

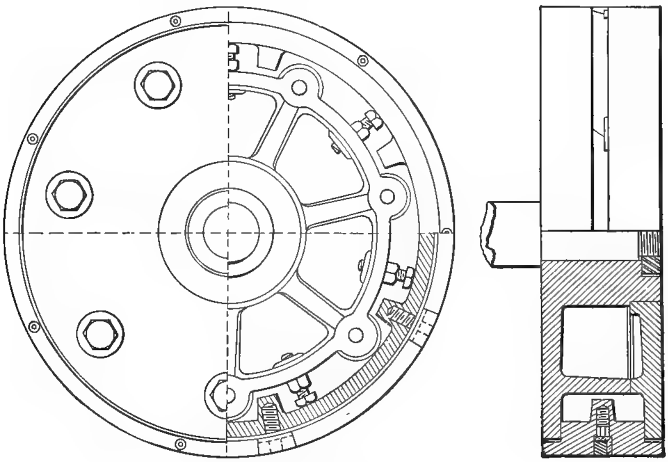


Fig. 172.—Harris piston.

the counterbore at each end of the cylinder, so that no shoulders can be worn on its surface.

In this design of piston the packing is self-acting in its adjustment, while the adjustment for alignment can be readily made by removing the follower and setting up the screws provided for this purpose.

In Fig. 172 are shown a plan and section of the Harris piston, consisting of a seven-part spider and as many locked set-screws for central adjustment. The rings are also in seven spliced segments, set out with helical springs.

As the piston, together with the piston-packing, is one of the most important parts of the engine, particular attention has been given to its design and construction.

It is forced upon the rod by hydraulic pressure, which in the

larger sizes the bearing is made the full thickness of the piston, with a steel collar screwed on the end of the rod; otherwise a recessed nut, as shown.

The bull-ring extends over the full width of the piston, overlapping the piston and the follower. It is grooved to receive the segmental packing.

All of the Harris pistons are fitted with bronze adjusting-screws, so that they can be kept centrally within the cylinder and that the rod may always be in perfect alignment.

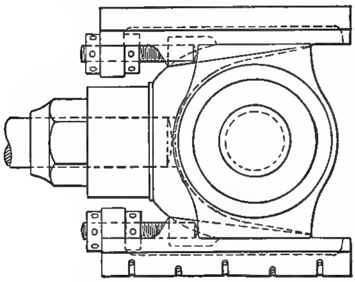


Fig. 173.—Nordberg cross-head.

In Fig. 173 is shown a cross-head of the Nordberg engine, which is adjusted by a top and bottom wedge and screws with lock-nuts. The adjustment of the cross-head for keeping the piston-rod in the central line and parallel with the piston-bore is an essential part of engine-management; and these designs are most numerous, but all having or seeking the required ideal of perfect control and fixedness of the adjustment.

In Fig. 174 is shown a similar arrangement in the cross-head of the Murray engine, with cylindrical Babbitted bearings on wedge-shoes that are bolted to the main block.

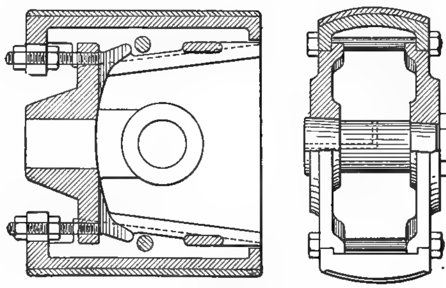


Fig. 174.—Cylindrical cross-head.

The design of the connecting-rods and the method of adjustment of their boxes vary very much with engine-builders, the wedge and screw being in general use.

In Fig. 175 is shown the rod-end used by the W. A. Harris Company, in which the inner box has an inclined back with a wedge and draw-screws on each side. The screw sprevent a possible movement of the wedge by the motion of the rod.

In Fig. 176 is shown the connecting-rod end of the Filer & Stowell Co., on which the adjusting-wedge is placed horizontally at both ends

of the rod, and adjusted by a collar-bolt and lock-nuts. A set-screw underneath is added as a safety-check.

The main bearings of a steam-engine require the same care as its other running parts—not only in providing for its proper lubrication, but also as the means for taking up of the boxes to meet the wear.

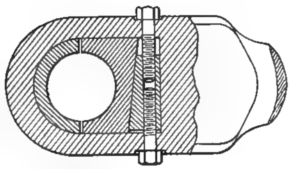


FIG. 175.—Cross-key box.

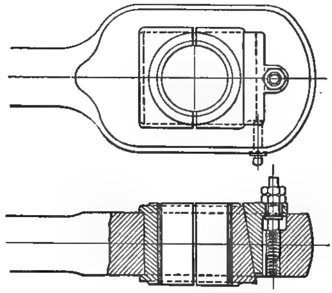


FIG. 176.—Side-key box.

In horizontal engines the thrust of the piston may cause a pound in the main bearing by looseness from wear, and in vertical engines the wear in the journal-boxes is vertical from the action of the piston, but may also be crosswise from the belt-pull.

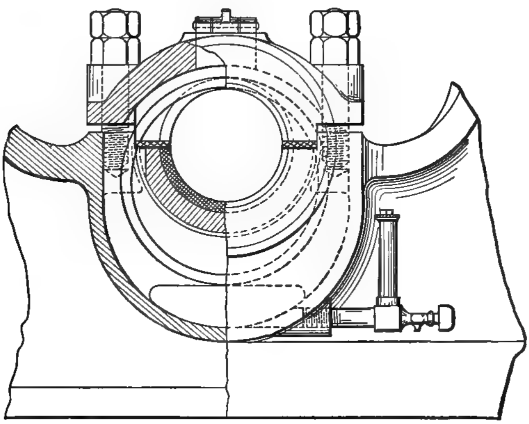


FIG. 177.—Main bearing, Bayley engine.

In Fig. 177 are shown a half-view and half-section of the self-lubricating main bearing of the Bayley engine, in which a ring dips into the oil-chamber below, and, rolling over the top of the journal, gives it a constant and economical oil-feed. A gauge-glass connected

to the bottom of the chamber shows the height of the oil, and the drip-cock allows of drawing off the oil and cleaning the chamber when required.

The adjustable main bearing used on the Todd engine is shown in section in Fig. 178, in which the quarter-boxes on each side are ad-

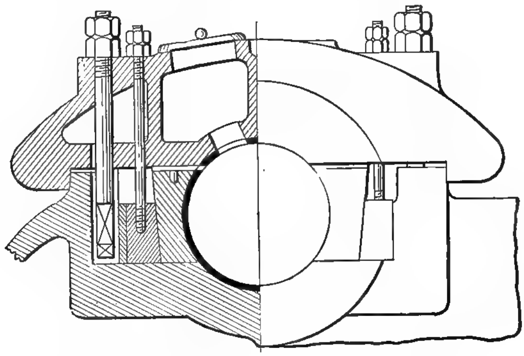


FIG. 178.—Main bearing, Todd engine.

justed by wedges at their back and by long stud-bolts reaching through the cap with locking-nuts. The cap of this bearing has a hand-hole and cover large enough to allow the journal-surface to be examined while the engine is running.

THE FLY-WHEEL

The most important element in controlling the motion of a steam-engine is that which equalizes its transmission of speed. The fly-wheel takes action before the governor in all motors or engines that receive their impulse in unequal increments, and the more unequal the impulse the more important is the office of the fly-wheel.

Its power to equalize the speed of revolution depends upon its weight and rim-velocity, while the governor only regulates the impellent force that drives the engine. Solid fly-wheels of cast iron as ordinarily made are limited to a rim-speed of about one mile per minute, with their usual diameter limited at 8 feet, and from 8 to 16 feet in halves, divided through the arms or between them. Above 16 feet in diameter sectional wheels are of the usual construction, with the joints at the end of the arms, and sometimes with the arm and section cast in one piece.

Fig. 179 shows a channel rim-wheel, as usual made in halves, and Fig. 180 a heavy rim-wheel in sections bolted to the arms.

The ultimate strength-efficiency of the wheel divided in halves, with ordinary bolting at hub and rim, is about 25 per cent., the sec-

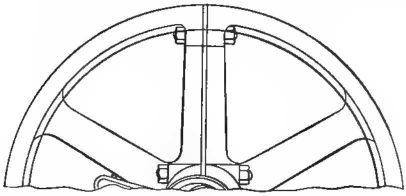


Fig. 179.—Wheel in halves.

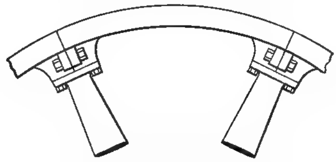


Fig. 180.—Sectional wheel.

tional about 50 per cent., and with heavy rim-wheels reënforced with links shrunk on, an efficiency of 60 per cent. may be obtained.

In the following table are given the number of revolutions of solid wheels and the different models of sectional wheels according to their percentage of efficiency, with a speed-margin of safety of one-third of their tensile strength.

This table has been computed on the assumption of 100 feet per second rim-speed as the maximum safe velocity of the rims of solid cast-iron wheels.

For sectional wheels with flange-joints between the arms, $100 \times \sqrt{.25} = 50$ feet per second.

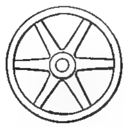
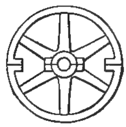
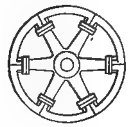
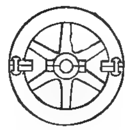
For sectional and belt wheels having joints at the end of the arms and rim-sections, $100 \times \sqrt{.50} = 70.7$ feet per second.

For thick-rimmed sectional wheels having rim-joints reënforced by steel links shrunk on, $100 \times \sqrt{.60} = 77.5$ feet per second.

From this may be deduced the number of revolutions for any diameter of the four conditions of efficiency: For solid wheels, 1,910 ÷ diameter in feet; for sectional belt-wheels and split wheels with joints between the arms, 955 ÷ diameter in feet; for sectional belt-wheels with additional joints at end of arms, 1,350 ÷ diameter in feet; for thick-rimmed sectional wheels having rim-joints reënforced by steel links shrunk on, 1,480 ÷ diameter in feet.

The weight and diameter of a fly-wheel are matters of much consideration in their design to meet the most economical conditions of the variable forces due to impulse, local limitations, and weight of metal to satisfy the requirement.

TABLE XXXII.—SAFE SPEED FOR CAST-IRON FLY-WHEELS.

	Efficiency of Rim-Joint.			
	1.00	.25	.50	.60
				
Diameter in feet.	R. P. M.	R. P. M.	R. P. M.	R. P. M.
1	1,910	955	1,350	1,480
2	955	478	675	740
3	637	318	450	493
4	478	239	338	370
5	382	191	270	296
6	318	159	225	247
7	273	136	193	212
8	239	119	169	185
9	212	106	150	164
10	191	96	135	148
11	174	87	123	135
12	159	80	113	124
13	147	73	104	114
14	136	68	96	106
15	128	64	90	99
16	120	60	84	92
17	112	56	79	87
18	106	53	75	82
19	100	50	71	78
20	95	48	68	74
21	91	46	65	70
22	87	44	62	67
23	84	42	59	64
24	80	40	56	62
25	76	38	54	59
26	74	37	52	57
27	71	35	50	55
28	68	34	48	53
29	66	33	47	51
30	64	32	45	49

For automatic valve-geared engines we quote Professor Thurston's general rule as the means of practice to meet the special conditions of engine-running:

Weight of fly-wheel $= 250,000 \frac{A\ S\ p}{R^2\ D^2}$, in which A = area of piston in square inches; S = stroke in feet; p = mean effective pressure; R = revolutions per minute; D = outside diameter of fly-wheel in feet.

As rim-velocity increases as the diameter, and as centrifugal force increases as the square of the rim-velocity, the centrifugal force must bear a safe proportion to the minimum tensile strength of the rim for assigning the weight and diameter of a fly-wheel for any assumed speed.

The total strain in a fly-wheel rim by centrifugal force is: $\frac{W\ V^2}{g\ R}$, in which W = the total weight of the rim in pounds; V^2 = the square of the velocity of the rim; g = the gravity 32.16, and R = the radius of the wheel in feet.

For example: a wheel-rim 1 inch square and 12 inches outside diameter would have a mean diameter of 11 inches $\times$ 3.14 = 34.54 cubic inches; at .25 pounds per cubic inch = 8.64 pounds. At a speed of 1,910 revolutions per minute, 1,910 $\times$ 3.1416 = 6,000 feet per minute, or 100 feet per second. Then $\frac{8.64 \times 10{,}000}{32.16 \times .5\text{ ft.}} = 5{,}360$ pounds, the total strain due to centrifugal force; and as there are two sides to the rim on which this force is divided, then $\frac{5{,}360}{2} = 2{,}680$ pounds strain per square inch, or about one-half the elastic limit and one-fourth of the lowest tensile strength of cast iron.

THE CONNECTING-ROD ANGLE

The position of the piston in parts of its stroke does not coincide with its position in parts of the crank-rotation, as will be seen by comparing the scales A and B, Fig. 181, made from the angular positions of a connecting-rod of twice the stroke in length. The upper scale, A, is in equal parts of the piston-stroke, while the lower scale, B,

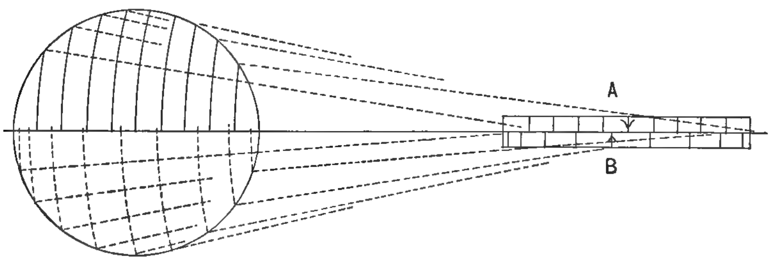


FIG. 181.—Piston- and crank-stroke.

represents the corresponding scale for equal increments of the crank-pin half-circle for one stroke. It may be noted by the scale that at the half-stroke of the crank the piston has advanced about $\frac{5}{8}$ of $\frac{1}{10}$ of its stroke, and that in the return-stroke the advance will be the same, making $1\frac{1}{4}$ of $\frac{1}{10}$ of its stroke difference in the positions of the piston during a revolution. The displacement of the position of the piston at half-crank stroke may be readily computed—say, for a 10-inch stroke and 20-inch piston-rod—by the right-angle equation, as follows: $20^2-5^2=375$, and $\sqrt{375}=19.365$, and $20-19.365=.625$, or $\frac{5}{8}$ inch, or a difference of $1\frac{1}{4}$ inches in the forward and back stroke of the piston. Short eccentric-rods produce the same irregular motion of the valve in a small degree by advancing its position at the middle of each stroke.

There is much in steam-engine design that cannot be detailed in a general treatise as shown in this work.

Experience is the most important factor in the details of construction that a designer should have in order to accomplish good results from the theory and in the line of modern practice.

The foregoing chapter is characteristic of the views of authors on the proportions of the leading parts in construction design; they may vary somewhat from the most recent practice of builders, but will make a good study for the inquiring engineer and the student in steam-engine design.

CHAPTER XV

THE SLIDE-VALVE AND VALVE-GEAR

THE slide-valve of our forefathers was the so-called D valve, without lap or lead, which has long since been relegated as a memorial of primitive experience. The D valve and its congener, the piston-valve, as we now understand them, are modelled after our modern ideas of the economy derived from expansion, compression, and steam-lead. For this purpose they are designed with extensions of their faces for steam- and exhaust-lap, and operated by the angular position and throw of the eccentric for steam- and exhaust-lead; all of which are made variable to meet the special contingencies of design in engine-economics.

In Fig. 182 we illustrate the modern type of the D valve with steam- and exhaust-lap, and in its central position.

The portion a is the "steam-lap" or "outside lap," and the portion b the "exhaust-lap" or "inside lap," and in order to open either port to steam or exhaust, it is necessary for the valve to travel from mid-position a distance equal to these laps, and for a full port-opening a greater distance than the amount of all the laps, or equal to the steam-lap and port width.

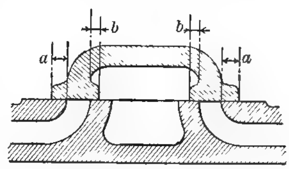


FIG. 182.—D valve.

In the ordinary type of plain slide-valve engine with throttle-valve the cut-off is fixed, and there is no adjustment for speed variation except through a governor and throttle-valve; while the automatic high-speed engines are provided with fly-wheel governors which vary the throw of the eccentric and the cut-off.

Since the valve-travel depends upon the lap and lead, both of which are frequently adjustable irrespective of the eccentric, the conditions in the cylinder also depend upon these quantities, and when the latter are determined the eccentric must operate the valves so as to produce the required lead.

The operation of the valve is determined by inspecting or meas-

uring its movements, the eccentric thus far having no direct influence upon the result. When the lap and lead are decided upon, the eccentric is turned until the desired lead is obtained, which is measured at the valve and not at the eccentric. As the lead with a given lap depends more or less upon the lap and upon lost motion in the gear, it is evident that these quantities must first be determined and the eccentric moved backward or forward on the shaft until the desired movement of the valve is obtained.

When the required steam-distribution has thus been established the eccentric will occupy the proper position for producing the results, and it will be seen that it makes absolutely no difference to the engineer whether the eccentric leads the crank by 98 or 105 or any other number of degrees. The position of the eccentric in cases of valve-setting and gear-adjustment takes care of itself by its movement to meet the valve-adjustment.

It will no doubt be apparent that the truly important point to be observed is the effect of lap and lead on the steam-distribution. When these have been properly determined and the movements of the valve regulated and timed to produce them, the eccentric will be found to be located at precisely the proper angle with reference to the crank.

SIMPLE SLIDE-VALVE GEAR

The relative position and motion of the eccentric-rod and valve-rod are points of consideration in the design of a plain slide-valve engine.

In determining the position of the eccentric in cases where a rocker-arm intervenes between the eccentric- and valve-rods, con-

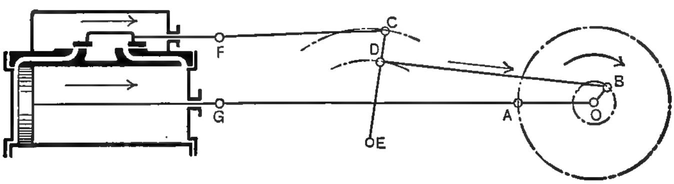


FIG. 183.—Direct gear for running "over."

sideration must be given the point at which the rocker-arm is pivoted. If the rocker-arm is pivoted so that the valve- and eccentric-rods

both move in the same direction at the same time, the eccentric is set in the positions shown in Figs. 183 and 184, which illustrate respectively the different parts of the gear in the proper relative positions when it is desired to have the engine run "over" and "under."

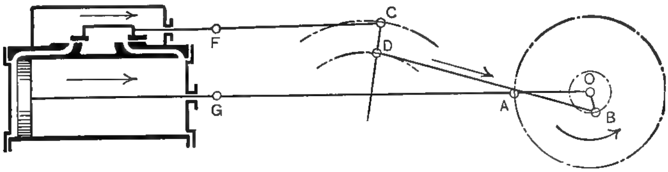


FIG. 184.—Direct gear for running "under."

When the rocker-arm is pivoted so that the valve- and eccentric-rods move in opposite directions, then the eccentric must be set in the positions shown in Figs. 185 and 186, which are directly opposite to the positions shown in Figs. 183 and 184. Here, too, the illustrations show the engine set to run "over" and "under."

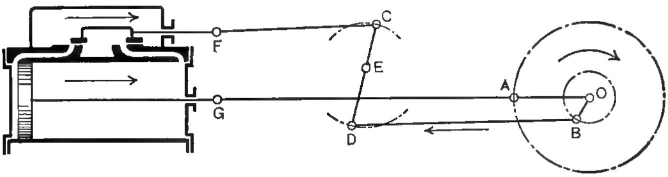


FIG. 185.—Indirect gear for running "over."

The eccentric-rod may be said to be indirect-acting, for with the rocker-arm pivoted as shown in the diagram the eccentric-rod has a movement that is directly opposite to that of the valve-rod.

This will be made plain by a study of the diagrams shown.

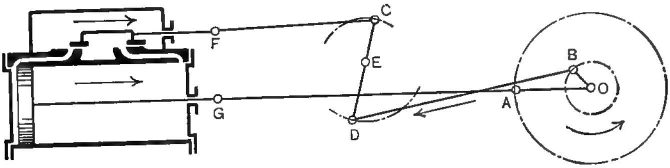


FIG. 186.—Indirect gear for running "under."

Fig. 184 shows the correct position of the eccentric when the engine is to run "under" and when the particular valve-gear shown in Fig. 183 is used.

In the diagrams, O represents the crank-shaft, A the crank-pin, C and D the points at which the valve- and eccentric-rods are connected to the rocker-arm, E the pivot for the rocker-arm, and F and G the necessary joints for the valve-stems and piston-rods.

A good rule to bear in mind when setting engine-valves that are operated by these simple types of valve-gear is: When a slide-valve is used and the eccentric- and valve-rods move in the same direction, set the eccentric ahead of the crank and make the angle between them 90 degrees plus the angle of advance.

The above rule holds good whether the engine runs "under" or "over." On the other hand, when the valve- and eccentric-rods move in opposite directions, the eccentric must be set behind the crank, and the angle between them will be 90 degrees minus the angle of advance.

The proper positions of the valve must be determined at the valve, whether the position of the eccentric be located first or last; but when observing the valve first, no attention need be paid to the position of the eccentric.

Few would care to undertake the work of placing the centre of the eccentric at a given angle to the centre of the crank, and no matter how carefully the work may be done, an inspection of the valve, in nine cases out of ten, will prove the time and labor to have been expended uselessly.

The length of the port-opening is usually equal to or little less than the cylinder diameter. The width of the bridge should be amply sufficient to prevent steam leaking past the seal into the exhaust-port or to prevent the over-travel uncovering the steam- and exhaust-ports at the same time.

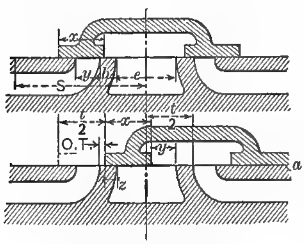


FIG. 187.—Middle and extreme valve-positions.

In order that the exhaust-port shall at no time be contracted to a less area than that of the steam-port, its width should be such that it will at all times retain an opening under the valve equal to the area of the steam-port at least.

In Fig. 187 is shown a good-proportioned D slide-valve at middle and extreme travels with the exhaust-port of a width more than double the width of the steam-

port, which allows of excess of valve-travel without restricting the exhaust-opening.

Such a valve, having an outside lap of $\frac{1}{2}$ inch, inside lap of $\frac{1}{8}$ inch, over-travel of $\frac{3}{8}$ inch, with ports and bridges 1 inch each, will have a travel of $\frac{1}{2}+\frac{1}{8}+\frac{3}{8}+1\times 2=4$ inches.

In laying out a valve it often happens that in order to have the points of release and compression occur at a particular period the inside or exhaust lap becomes zero, or else leaves the exhaust-port slightly open at mid-position. This port-opening is known as *negative* lap, and it is a common occurrence to find a valve possessing negative lap when not so designed, due to the effect of the rod-angularity which was neglected in the valve-diagram. This only occurs, however, when the original exhaust-lap is very small, so that the small error caused by the angularity of the rod neutralizes the shortage in exhaust-lap entirely.

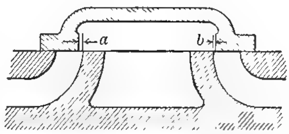


FIG. 188.—Negative inside lap.

The zero and negative laps are shown in Fig. 188, in which a is the point of opening of the exhaust, and b the point of cut-off for the exhaust.

The lead to be given to a valve depends largely upon the style of engine, and must be governed entirely by experience. It may vary from 0 up to and even above $\frac{3}{8}$ inch. Slow-running engines require less lead than those running at a high speed, and engines having a high compression require less than those without, since the clearance-space is filled with compressed steam whose pressure is nearly equal to that of the entering live steam. In locomotives it varies from 0 to $\frac{1}{2}$ inch, and in marine practice from 0 to $1\frac{1}{2}$ inches. In stationary practice the values may vary from 0 to $\frac{1}{2}$ inch, but seldom more, unless in very high-speed work. The angle of lead, which is the angle that the crank makes with the dead points at admission, varies between 0 and 8 degrees in stationary practice, and seldom over 10 degrees in marine practice.

The inside lead, or what would be called the exhaust-lead, is often greater. It is provided for the purpose of opening the exhaust-port early. Weisbach gives the proportion of $\frac{1}{25}$ to $\frac{1}{15}$ of the valve-travel.

The outside lap may vary from $\frac{1}{4}$ inch to $1\frac{1}{4}$ inches in locomotives,

while in marine engines it may run from $\frac{1}{2}$ inch to $3\frac{1}{2}$ inches. The inside lap may vary from a negative value to $\frac{1}{4}$ inch, and in marine practice will often run as high as $1\frac{1}{2}$ inches.

The eccentricity will vary largely with the style of engine and valve-travel desired, but it should be no greater than necessary, owing to the wear on the reciprocating parts. In marine engines it may vary from 5 to 8 inches, or even more in large units.

Changing the dimensions of any part of the valve which determines either the lap or lead, or changing the angular advance, alters the steam-distribution. The accompanying table shows at a glance just what particular effect each change has upon this distribution:

TABLE XXXIII.—EFFECT OF CHANGING OUTSIDE LAP, TRAVEL, AND ANGULAR ADVANCE. *Thurston.*

CHANGE.	ADMISSION.	EXPANSION.	EXHAUST.	COMPRESSION.
Increase of outside lap.	Begins later. Ceases sooner.	Occurs earlier. Continues longer.	Unchanged.	Unchanged.
Decrease of outside lap.	Begins earlier. Ceases later.	Begins later. Period shortened.	Unchanged.	Unchanged.
Increase of inside lap.	Unchanged.	Begins as before. Continues longer.	Begins later. Ceases earlier.	Begins sooner. Continues longer.
Decrease of inside lap.	Unchanged.	Begins as before. Period shortened.	Begins later. Ceases earlier.	Begins later. Period shortened.
Increase of travel.	Begins sooner. Ceases later.	Begins later. Ceases sooner.	Begins later. Ceases later.	Begins later. Ends sooner.
Decrease of travel.	Begins later. Ceases earlier.	Begins earlier. Ceases later.	Begins earlier. Ceases earlier.	Begins earlier. Ceases later.
Increase of angular advance.	Begins earlier. Period unchanged.	Begins sooner. Period unchanged.	Begins earlier. Period unchanged.	Begins earlier. Period unchanged.
Decrease of angular advance.	Begins later. Period unchanged.	Begins later. Period unchanged.	Begins later. Period unchanged.	Begins later. Period unchanged.

EXCESSIVE COMPRESSION

The compression is sometimes represented in indicator-diagrams as excessive; with plain D valves its relief cannot always be found in the valve-gear adjustment. A method of changing the exhaust-lap

has been proposed, and is shown in Fig. 189. It consists in filing off the edge of the lap for small amounts, and for a further change filing half-round grooves across the edge of the lap and the port at opposite points. This method of reducing compression will slightly interfere with the expansion-line, but not to any serious extent.

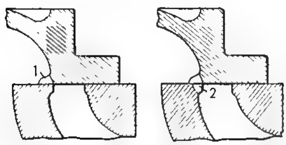


FIG. 189.—Changing the exhaust-lap.

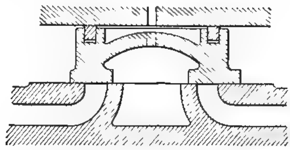


FIG. 190.—Balanced valve.

The modifications of the D valve for more perfect action have developed some curious yet valuable features in their design.

The balanced valve is now largely in use, and one of its forms is shown in Fig. 190, which carries in the back a ring which bears against a smooth seat on the valve-chest cover and is supported by springs. The cavity at the back of the valve may be open to a condenser or through the back of the valve to the exhaust.

As one representative of the double-admission type, the Allen balanced valve, illustrated in Fig. 191, has a broad, double-ported steam-passage through the body and a relief-port from the cavity at its back to the exhaust-cavity. The ports of the supplementary passage are so located as to take steam to the port at the moment of

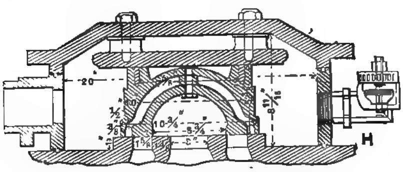


FIG. 191.—Allen balanced valve.

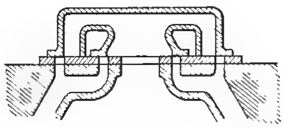


FIG. 192.—Double-ported slide-valve.

opening of the lap-edge of the valve, from the simultaneous opening of the supplementary port at the other end. This passage never communicates with the exhaust, for its outlet to the main port is closed just before the port opens for release, and is opened just after the port is closed for the exhaust. Its economy lies in its short travel. The double-ported marine slide-valve, shown in Fig. 192, is another

novelty in the line of double ports. It is in use on marine engines on the intermediate and low-pressure cylinders.

The valve is shown in its middle position, in which all the ports are opened and closed alike and with but slight cut-off. The short travel required by double ports makes this type of valve desirable in the triple and quadruple engines.

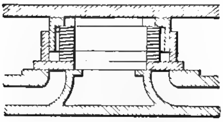


Fig. 193.—Balanced slide-valve.

A type of balanced slide-valve much in use is shown in section in Fig. 193, consisting of a ring held in a recess at the back of the valve and pushed against the steam-chest cover by springs. The details of its construction vary somewhat in the different engines to which it is applied. The relief-pressure varies from 60 to 80 per cent.

In Fig. 194 are shown the face and back of a balanced slide-valve and the steam-chest, with the valve in position as used on the Skinner high-speed engine.

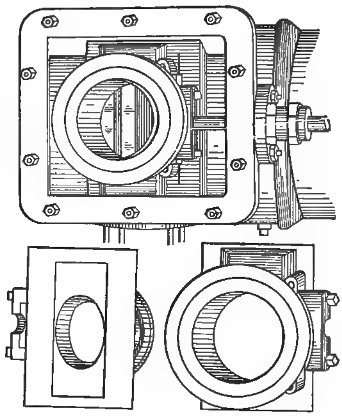


Fig. 194.—Balanced slide-valve, Skinner model.

The valve has 80 per cent. of its area relieved of pressure by a balance-ring which rides against the steam-chest cover. This ring is free to revolve, and changes position with every stroke of the valve, preventing any creasing or cutting of seat, ring, or cover. Steam packing-rings prevent leakage between the ring and the hub of the valve. This construction allows a large port-area, which is necessary for proper steam-distribution. Twenty per cent. of steam-pressure is sufficient to hold the valve in steam-tight contact with the seat and to take up the wear. The valve is free to lift from the seat in case water enters the cylinder.

SLIDE-VALVES WITH A RIDING COVER

Of this type of slide-valve there are many designs, with both single and double ports so arranged as to facilitate a full passage of steam with the shortest travel of the valve. In Fig. 195 is shown

a section of the cylinder and steam-chest of the Ames engine. The valve is a two-ported plate, one at each end, riding under a partially balanced pressure-plate, with recesses to allow of double-port openings and a short valve-travel. The supplementary port-opening is over each end of the valve-plate, as shown. This type is representative of a large number of engine slide-valves by different builders.

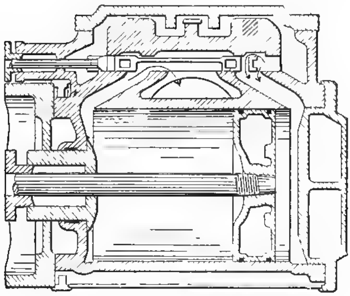


FIG. 195.—Ames slide-valve.

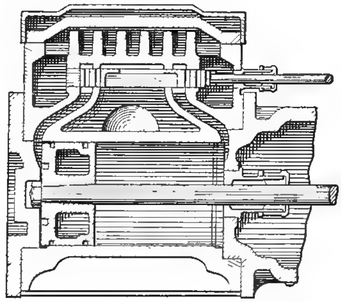


FIG. 196.—Chandler & Taylor double-ported slide-valve.

In Fig. 196 is shown a section of the cylinder and steam-chest of the Chandler & Taylor engine—a high-speed model designed with special adaptation for direct electric generator-connection.

The valve-plate is two-ported at each end, a feature making it possible to give a large port-area with short valve-travel. This class of valves is so balanced and free to lift that there is little or no danger from slight excess of water in the cylinder.

THE SLIDE-VALVE WITH INDEPENDENT CUT-OFF

The most economical use of steam requires an earlier cut-off than can be obtained from the single D valve and a proper adjustment of the exhaust- and compression-lines.

In order to obtain the short cut-off with the required conditions of exhaust and compression in the slide-valve, a number of devices have been proposed and used in Europe and the United States. Among such are the Gonzenbach, with a three-ported solid rider in a separate steam-chest at the back of the regular D valve; the solid sliding valve on the back of the main valve, as designed by Breval,

Polonceau, Napier & Rankin, Farcot, Borsig, A. K. Rider, and other models, most of which are used in locomotive service.

The Meyer expansion-valve has proved its value for stationary service by its continued use for more than a half-century. In this valve the riding cut-off consists of two blocks adjusted for any required cut-off by right- and left-handed screws on a traversing spindle, as shown in Fig. 197. A, A are the cut-off blocks adjusted by the right and left threads on the spindle B, which extends through both ends of the steam-chest, with a swivel at G, a wheel, H, for turning the spindle, and an index at i to designate the amount of cut-off; S is the half-travel of cut-off valve. At the right is a diagram of the eccentrics and crank-pin. The lap and lead of the main valve are not effected by the operation or adjustment of the cut-off valve.

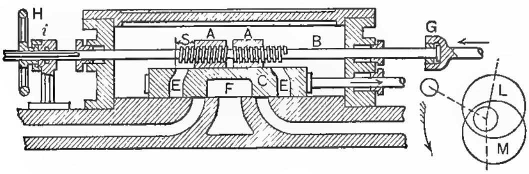


Fig. 197.—Meyer expansion-valve.

A novel arrangement of balanced valves with simultaneous movement for both cylinders of a compound tandem type, as used on the

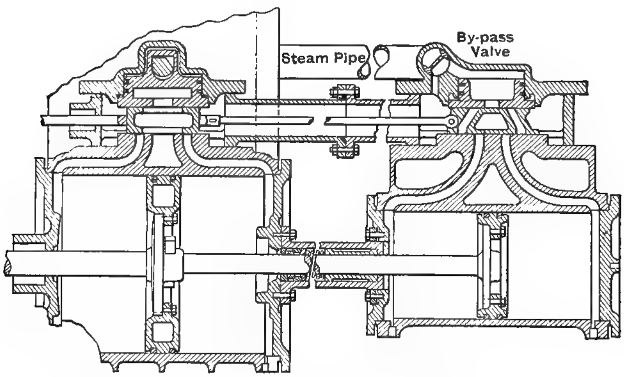


Fig. 198.—Union valves, compound tandem engine.

locomotives of the Pittsburg Locomotive Works, is shown in Fig. 198. The cylinders have a sleeve between the heads which carries the piston-rod without packing.

The valves are connected by a rod passing through a pipe between the steam-chests of the high-pressure and low-pressure cylinders. The high-pressure valve receives steam through the balance-plate, which is movable, with piston-rings for perfect closure. In ordinary use the exhaust-steam from the high-pressure cylinder passes through the pipe covering the connecting-rod to the steam-chest of the low-pressure cylinder. A by-pass valve and side port allow of turning high-pressure steam directly to the low-pressure cylinder when needed for starting.

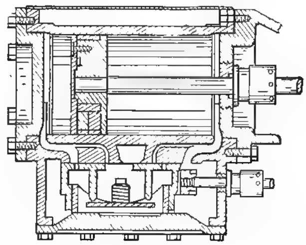


FIG. 199.—Brownell balanced slide-valve.

In Fig. 199 is shown a section of the cylinder and balanced valve of the Brownell engine. The valve is of the box type, double-ported for both steam and exhaust, and practically perfectly balanced.

The steam-pressure is removed from the back by means of a balance-ring which bears against the steam-chest cover. A coil-spring serves to keep the ring against the chest-cover, thus taking up the wear automatically and preventing the ring from leaving its seat and causing annoyance by rattling. This class of balanced valves is in use on the Skinner, Ball & Wood, Payne, Erie, and other high-speed engines.

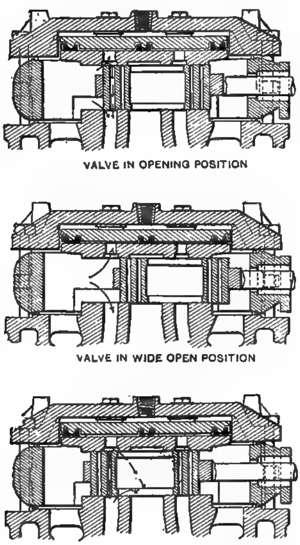


FIG. 200.—Wilson slide-valve.

A balanced valve of the Wilson type is shown in three positions in Fig. 200. The uppermost figure shows the opening position of the double port; the middle figure, the wide-open position, and the lowermost figure, the exhaust position. The gridiron form of the valve shortens the valve-travel and doubles the port-area. The exhaust-ports are also double in the action of this valve. The ported balance-plate rests under an adjusting-plate bolted to the steam-chest cover, which makes the valve almost frictionless.

In Fig. 201 is shown the balanced slide-valve of the Bayley vertical automatic engine. The pressure-plate is free from the steam-chest cover and pressed against the valve by a spring; it is held in place by adjustable stays against the ends of the steam-chest and by side-bars to prevent lateral motion.

The oscillating cylindrical valve, shown in Fig. 202, is still much in use on hoisting-engines with cam, eccentric, or secondary crank-motion. It seems to be well adapted to the operation of hoisting and other small engines by the simplicity and direct action of its gear. Steam enters at the top of the cylinder at S and passes around the cylinder to the valve-chest at P, P—a means for keeping the cylinder clear of water while

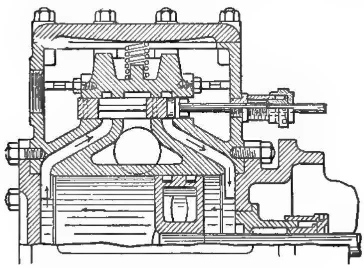


Fig. 201.—Bayley slide-valve.

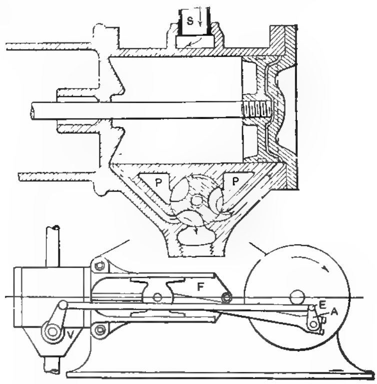


Fig. 202.—Oscillating valve.

the hoist is waiting. The valve-arm V is directly connected to the crank-pin arm at E. These valves operate on the same principle as the plain slide-valve, with from five-eighths to three-quarters cut-off for light work, or full stroke for heavy, slow pull and two cylinders.

The gridiron or multiported valve is much in use in the marine service and on the stationary engines of the Slater Engine Company, McIntosh & Seymour, the American & British Mfg. Co., and C. H. Brown & Co.

In Fig. 203 is shown a cross-section of the cylinder of the McIntosh & Seymour engine, with the valve-gear of the steam, exhaust, and cut-off valves. The four valves, steam and exhaust, are operated from a rock-shaft at M, which in turn is rocked by a fixed eccentric through a bell-crank lever connected to a crank on the rock-shaft. The oscillating pin P, on the rock-shaft wrist-plate at M, operates the exhaust-

valve by a direct link and cross-head, c, while the steam-valve is operated from the pin P′ through the link-rod S and toggle-joint mc. The riding cut-off valve is operated from a second rock-shaft located at A, with its variable motion controlled by the fly-wheel governor, which revolves the eccentric to advance the cut-off valve. The connection between the eccentric and rock-shaft is a bell-crank lever with a slipper-link bearing on the eccentric from one arm and a link to a crank-pin on the rocker-shaft.

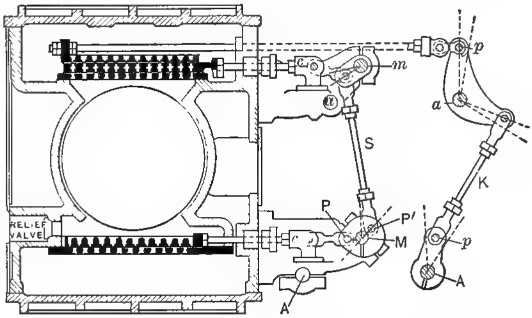


Fig. 203.—Gridiron valves and cut-off.

The connection between the rocker-shaft and the cut-off valve is shown in the figure at the right by a rocker-arm at A, linked to a bell-crank rocker pivoted at a, so arranged that the cut-off valve moves in the opposite direction to the main valve, and with the rapid closing of the ports giving a sharp corner on a card at the end of the admission-line.

In Fig. 204 is shown an enlarged section of the gridiron valve and riding cut-off grid of the McIntosh & Seymour engine.

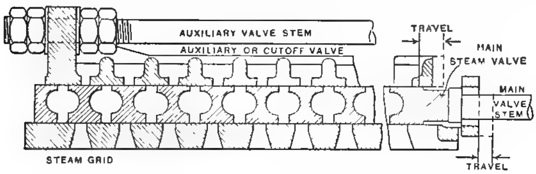


Fig. 204.—Section of gridiron valve and cut-off.

In Fig. 205 is shown a section of the steam and exhaust gridiron valves and valve-gear of the Brown engine, and in Fig. 206 the motion-gear of the exhaust-valve. The vertically operated steam-valves and steam-chest are on the side of the cylinder, while the exhaust-valves are horizontal and beneath the cylinder.

The operation of these valves may be understood from the cuts and reference-letters.

The lifter A, which is connected to the lower arm of the bell-crank

lever B, has just engaged the latch C, which is journalled on a pin on the guide D. When the long arm B is drawn toward the crank-shaft by the eccentric, the lifter A is raised, which carries the latch and guide up with it and causes the valves to open the ports. This upward movement continues until the outer end of the latch engages the trip-lever E, which causes the latch to let go of the lifter, when the valve immediately descends, dropping by its own weight, and being cushioned by the dash-pot directly under the guide, which movement closes the ports and effects the cut-off, which, owing to the short travel of the

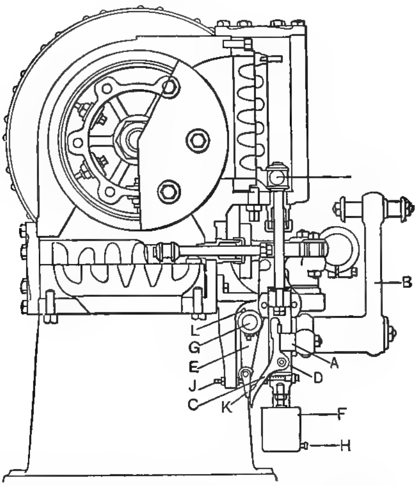


Fig. 205.—Gridiron valves, Brown engine.

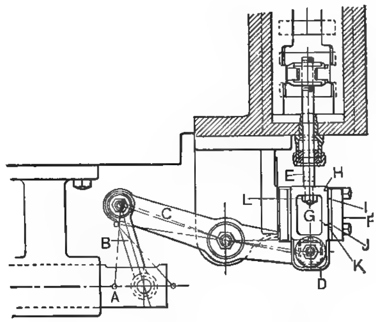


Fig. 206.—Exhaust valve-gear.

valve, is very sharp, permitting no wire-drawing of the steam. The trip-lever E is carried by the auxiliary shaft G, which is connected to and is actuated by the governor.

The exhaust-valve mechanism is shown in Fig. 206, which represents a plan. The exhaust-valves have a positive connection with the sliding bar A, and hence have an unvarying travel.

When the sliding bar A is moved to the right in the cut by the exhaust-eccentric, the longer arm C of the exhaust-lever is moved inward, while the shorter arm is moved outward, which opens the exhaust-ports, the reverse movement taking place when the port is closed.

THE DIAGRAM OF THE SLIDE-VALVE FOR CUT-OFF

The lay-out of a slide-valve diagram is an easy matter when once we consider its simple intricacies. Assuming that the lap is required and that the lead is prefixed to some definite amount between the usual variation from $\frac{1}{32}$ to $\frac{3}{16}$ inch or more; the lead in any case being kept as small as possible to allow the admission-line on the indicator-diagram to be vertical; to find the lap and lead.

A diagram for cutting off at three-quarters stroke is shown in Fig. 207, on a scale in parts of the engine-stroke, say for 12 inch, as shown in the diagram, for 12-inch stroke.

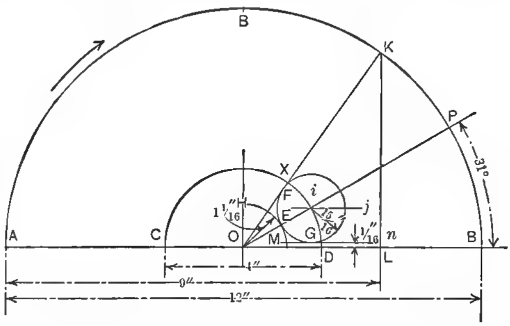


FIG. 207.—Lap- and lead-diagram, three-quarters cut-off.

In this diagram the connecting-rod angle is not considered, but should be allowed for in precise work, as shown in the foregoing chapter.

On the 12-inch base-line of the semicircle ABB measure the distance of the piston from A equal to the cut-off, say 9 inches, and from this point draw a vertical line to the semicircle at K; draw the line from the centre O to K, and the line Mn, at a distance from AB, equal to the lead, say $\frac{1}{16}$ inch. K is the position of the crank-pin at three-quarters stroke.

Next find the centre of a circle on the semicircle described by the centre of the eccentric, and describe the circle EFG, with its circumference tangent to the lines Mn and OK, as at F and G.

The radius of this circle will be the required lap to be added to the valve in order that it may cut off at three-quarters of the stroke. The amount of lap to be added for this point of cut-off is $\frac{15}{16}$ inch.

The diagram will also show the amount the port-opening has been decreased after adding lap to the valve. With O as a centre and the compass-pen open to a length equal to the distance OE, describe an arc of a circle that will be tangent to the lap-circle at E, and the

distance OE is the amount the port will be opened when the eccentric is at its greatest throw, which in this case is $1\frac{1}{16}$ inches.

By adding lap to the valve we have also changed the position of the centre of the eccentric.

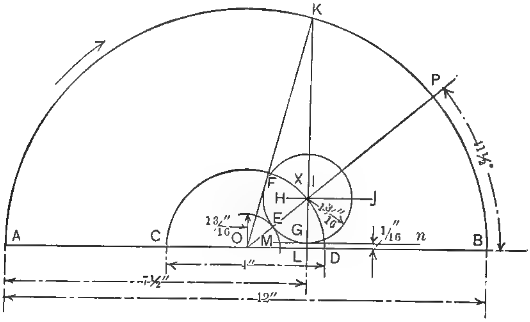


FIG. 208.—Lap- and lead-diagram, five-eighths cut-off.

Instead of being in a position of 90 degrees ahead of the crank, it must now be moved a considerable distance beyond this point and equal to the distance BP, or 31 degrees, or 121 degrees ahead of the crank.

In Fig. 208 is shown a diagram for five-eighths cut-off, with lead as before and lap of $1\frac{3}{16}$ inches, as shown, the port-opening being reduced to $\frac{13}{16}$ inch and the angle of advance being increased to $41\frac{1}{2}$ degrees, the total advance being $131\frac{1}{2}$ degrees.

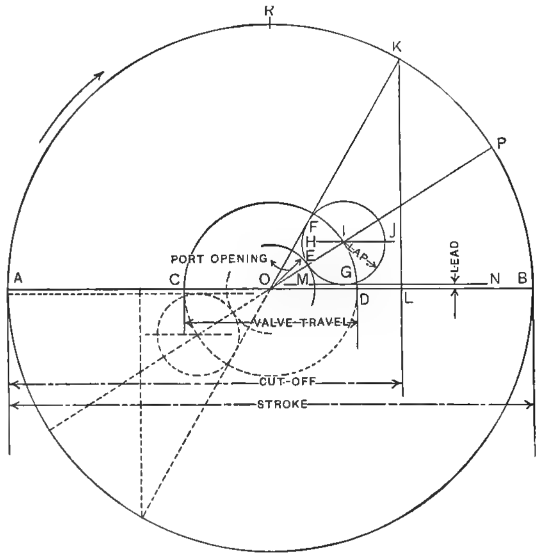


FIG. 209.—Universal valve-diagram.

Fig. 209 shows a diagram by which any position of the valve-gear and the different positions of the valve may be found when some of the other events are known. Suppose we know the valve-travel, the lap, and lead, and want to know what the cut-off and angle of advance will be. With the point O as a centre, and a radius equal to the length of the crank, describe the circle ARB. The diagram should be drawn to some scale, say 3 or 6 inches to the foot. The diameter AB of the circle will represent the stroke of the engine,

and the circle itself will represent the path of the centre of the crank-pin for one revolution.

Again, using O as a centre and a radius OD, equal to one-half travel of the valve, describe the semicircle CFD. The diameter CD will represent the travel of the valve.

Above and parallel to the line AB draw the line MN, a distance from AB equal to the known lead.

Draw also the line HJ parallel to the line MN, and a distance above it equal to the known lap.

The line HJ will intersect the semicircle CFD at I, and with I as a centre and a radius equal to the distance IG, which is the required lap, describe the circle FEG.

From the centre O draw a diagonal line that will be tangent to this circle at F and will intersect the large circle at K.

Next draw the diagonal line from the centre O and through I of the lap-circle, and intersect the large circumference at P. K will be the position of the crank-pin, and L will be the position of the piston in the cylinder when the steam is cut off. The angle BOP will be the required angle of advance.

Take another case, where we know the valve-travel, the angle of advance, and the cut-off, and we want to know the amount of lap and lead. Lay off the stroke the same as before, and draw the circumference described by the centre of the crank-pin, and suppose the piston and crank-pin to be in the same positions as before, at L and K. Draw the perpendicular line LK, and from the centre O draw the line OK. From B lay off the angle of advance, BOP. The line OP will intersect the semicircle at I. With I as a centre, draw a circle that will be tangent to the line OK, as at F. Draw the line MN, tangent to the circle just described and parallel to the line AB, and the distance between the lines AB and MN is the required lead, and the radius IG will be the required lap.

Take another case, where we know the lap and lead and point of cut-off, and wish to find the valve-travel and angle of advance. Proceed the same as in the previous cases, and locate the points K and L and also draw the line MN. Open the compass to the required lap and find by trial the centre I, from which a circle may be described tangent to the lines OK and MN, as at F and G.

Next, with OI as a radius and O as a centre, describe the semi-

circle CID, passing through I. The distance CD will be the travel of the valve. From O and through the centre I draw the line OP, and BOP will be the required angle of advance.

There is still one other case where this diagram can be used, where we know the point of cut-off, the lead, and the amount of port-opening when the valve is at the end of its travel, and we wish to find the lap, the valve-travel, and angle of advance.

The crank-pin and pistons are supposed to be in the same positions as in the previous cases, and the cut-off is also supposed to be the same.

Draw the lines OK and MN, and with O as a centre and a radius OE equal to the known port-opening, describe an arc of a circle. Next, by trial find the centre I of a circle, which, when described, will be tangent to the arc just drawn, and which will also be tangent to the lines OK and MN, as at F and G.

The radius of the circle will be the required lap. From O draw the line OP, and the distance BP will be the angle of advance. Through I draw a semicircle with a radius equal to OI. The diameter CD of this semicircle will be the distance travelled by the valve. In all of the above cases the radius OE represents the amount the port will be opened when the eccentric is on its dead-centre or at the position D. Notice also that the crank is supposed to be on the dead-centre at A, and that the engine is about to make the forward stroke. This makes it necessary to lay off the whole geometrical construction to the right of the centre O.

The measurements for the other stroke, with the engine on the dead-centre at B, will have to be constructed to the left of the centre O and below the line AB. The measurements for this stroke will be theoretically the same, and are shown by the dotted lines.

We could use the same diagram to lay out the movements for a piston-valve. A piston-valve, to cut off at the same point of the stroke, would require the same amount of lap; and if the nature of the work done by the engine was the same, it would require the same amount of lead.

A valve is said to be direct when it admits the steam to the cylinder past its outside edge, and allows the steam to exhaust to the atmosphere past its inside edge. A valve is said to be indirect when it admits the steam to the cylinder past its inside edge, and allows the exhaust-steam to escape past its outside edge.

These are the conditions of a piston-valve whose movement is just opposite to that of a slide-valve. Another thing to be noticed is the difference in the movement of the two valves relative to the movement of the piston. When the opening movement of the slide-valve occurs the valve and the piston both move in the same direction.

With the opening movement of a piston-valve the valve and piston move in opposite directions to one another. This makes it necessary to shift the centre of the eccentric to a position exactly opposite to that of a slide-valve.

THE PISTON-VALVE

The use of the piston-valve has been largely extended of late years by its advantages over the slide-valve in the accessibility of its parts, lightness, more perfect balance, and greater port-area, which features make it easier to handle, and decrease the wear and tear on the motion-work of an engine. With the increased size of engines and steam-pressure the ordinary D balance-valve increases in size proportionately, and while we may balance a slide-valve in the same ratio as the valves on smaller engines, the difference in the unbalanced surface increases with the size of the engine, and with it the wear on the valve, link-motion, and eccentric-straps, and the work necessary on the part of the engineer to handle the engine. This being a fact, a great deal of trouble is experienced in keeping the valves on slide-valve engines true to their seats, while on the other hand there is no trouble of this kind with the piston-valve until after the engine has been in use for a long while and the parts have become badly worn. The use of the inside admission piston-valve does away with the metallic valve-stem packing, which means a great saving, as there is only the exhaust-pressure on the packing side, and the fibrous packing answers the purpose and lasts a long while.

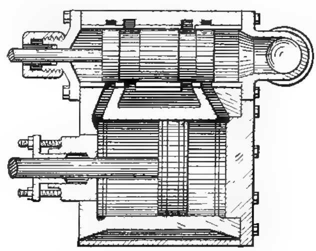


Fig. 210.—Noye piston-valve.

A simple piston-valve taking steam at its ends and by a passage through the valve is used on the Noye engines, and shown in Fig. 210. The valve has a long middle bearing and rides in a ported sleeve, forced

into the cylinder of which the steam-chest forms a solid part. The exhaust-ports are double, thus giving a long bearing to the valve. The valve-rod is connected direct to the eccentric and its rod by a ball-and-socket joint to correct any irregularity in the alignment.

Fig. 211 is a section of a simple hollow piston-valve, the rod of which passes through a central tube with bar-stays to the valve and a lining to the valve-chest. It can take steam at the centre or ends, as convenient, and operates exactly as a D slide-valve, with three times its extension of port-opening; hence short ports and valve-travel.

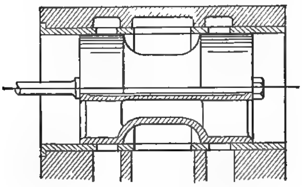


Fig. 211.—Hollow piston-valve.

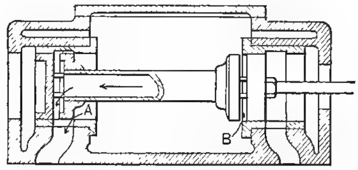


Fig. 212.—Armington & Sims double-ported piston-valve.

A double-ported piston-valve in use on the Armington & Sims engines is shown in section in Fig. 212. In this model the steam enters the steam-chest between the valve-pistons. It will be seen that steam-admission is through two ports at A and B in the valve and at both ends simultaneously. The steam entering at port B passes through the hollow neck of the valve, thus duplicating the port-opening for a short travel of the valve. For the exhaust the valve is single-ported, but the great width, due to its cylindrical form, gives ample port-opening with a short travel of the valve.

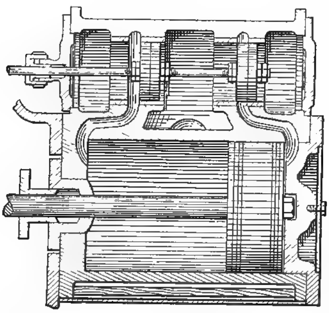


Fig. 213.—Harrisburg piston-valve.

In Fig. 213 is shown a section of the piston-valve chest and the valve used on the Harrisburg engine. The pistons of the valve are separate from the rod, though fastened to it and made adjustable by nuts and lock-nuts. It takes steam at the centre and exhausts from steam-chest ports outside the valve-disks. It has the same conditions as to lap and lead as the plain slide-valve,

but with the advantage that lap, both outside and inside, may be increased by changing or separating the parts of the disks with a washer-disk of any required thickness.

THE SLIDE-VALVE GEARAGE AND GOVERNORS

The plain slide-valve gear, as illustrated on page 225, is greatly modified and complicated for the various purposes of adjustment to suit the needed requirement for speed and reversing, and also for obtaining the valve-motion without the eccentric.

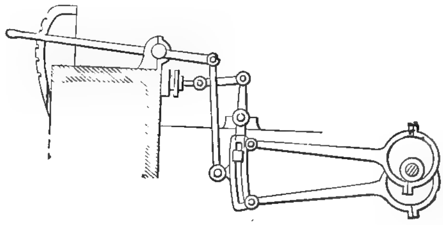


FIG. 214.—Stephenson link.

The link-motion from double eccentrics, much in use on marine engines and locomotives, is shown in Fig. 214. It has been long known as the Stephenson link. In this plan the slotted link is moved up or down for shifting the valve-motion or for reversing by a lever and connecting-rod. Its design has many forms to suit the varying conditions of this plan of link-motion, as shown in Fig. 215, and is largely in use in its simplest form on the engines of the smaller marine craft, from the pleasure-boat to the tug-boat, and on steam-driven automobiles and traction-engines. This link-motion is used in connection with ordinary D or gridiron valves, with the usual lap and lead for from five-eighths to three-quarters cut-off.

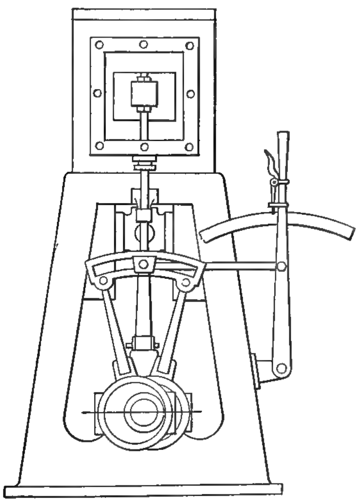


FIG. 215.—Link on vertical engine.

A reversing link-motion from a single eccentric is shown in Fig. 216, in which the slotted link is pivoted to the end of the eccentric-rod and is moved up and down by a bell-crank lever. The block carrying the valve-rod is stationary, allowing the pivoted link-centre to pass by it and thus reverse the valve-motion. There are

a number of curious models of valve-motion from single eccentrics, of which we select the following as examples of design in this interesting line.

A method of lengthening the stroke of an eccentric of small size by a link-connection to the eccentric-strap at a right angle with the

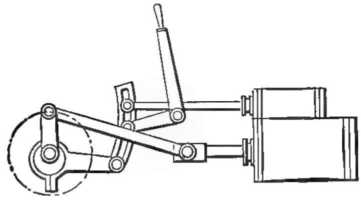


FIG. 216.—Single-eccentric reversing-gear.

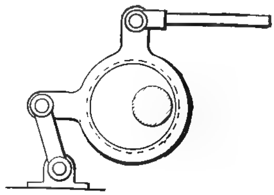


FIG. 217.—Method of increasing eccentric.

connecting-rod is shown in Fig. 217. In this way a considerable increase in the throw of the eccentric can be made, depending upon the relative lengths of strap-arms. A variable and adjustable throw of a single eccentric—called the Fink link-gear—for a D valve is shown in Fig. 218. A curved slot-link is made a fixture of the eccentric-strap, as shown, one end of which is pivoted to a swinging link attached to the engine-frame. The end of the jointed valve-rod is pivoted to a block in the link, and its distance regulated by a connecting-rod and screw pivoted to the valve-rod near the link.

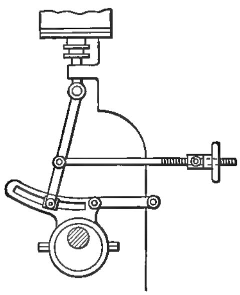


FIG. 218.—Direct variable valve-motion.

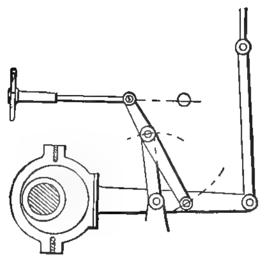


FIG. 219.—Variable link valve-motion.

Another and simple lever and link-movement for variable expansion from a single eccentric is shown in Fig. 219. The lever is pivoted to the connecting-rod of the eccentric and travels past the fixed pivot

of its link, thus swinging the end of the eccentric-rod up and down for the valve-motion. The position of the upper end of the lever is adjusted by a wheel and screw for varying the throw of the valve. Many variations of this idea have been proposed and used.

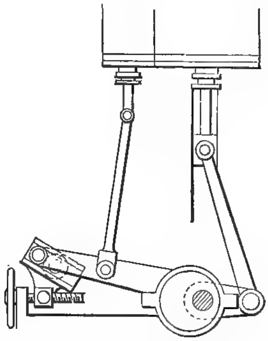


FIG. 220.—Block-link variable valve-motion.

Another design of past usage which is illustrated in Fig. 220, represents but one of the many strenuous efforts to utilize the single eccentric for the most efficient and economical work of the valve. Here the end of the eccentric-rod is pivoted to a block in a slotted link that by tilting up or down with a screw varies the throw of the valve, the valve-rod being pivoted to the eccentric-rod.

The Marshall valve-gear, shown in Fig. 221, is operated on the same general principles as the last two examples. A single eccentric, set opposite to the crank and its short connecting-rod, is pivoted to the valve-rod at J, with its end pivoted to a link, GF, which is also pivoted to a bell-crank lever at G, whose length, GH, is the same as that of the

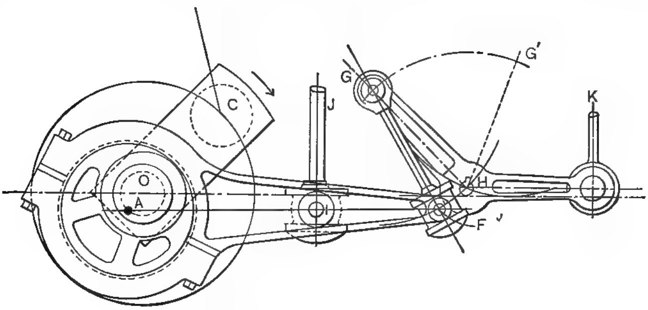


FIG. 221.—Marshall valve-gear.

link GF. The other end of the bell-crank lever is pivoted to the rod K and to a hand-gear for throwing over the bell-crank link to the position at G′ for reversing the engine, and to intermediate points between G and G′ for varying the cut-off. This gear gives a constant lead for both positions. The position of the gear in the cut is for running "over."

VALVE-GEARS WITHOUT ECCENTRICS

The idea of operating a steam-engine without an eccentric has been a theme of invention for a long time, and we illustrate those that have been in actual use and made a name for themselves—the Walschaert crank-pin arm and the Joy valve-gear.

One of the Walschaert valve-gears, which uses an eccentric, but modifies its link-movement by a cross-head link and lever for making the lead, is shown in Fig. 222. The eccentric-rod is pivoted to the lower end of a curved slotted link, itself pivoted at its centre to a fixed lug on the engine-frame. A bell-crank lever governs the position of the end of the valve-rod and its sliding block in the link. An arm from the cross-head is linked to a lever connected with the valve-rod and valve-stem for controlling the lead.

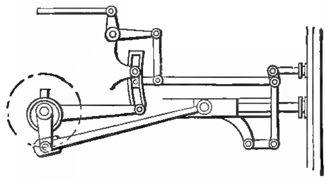


FIG. 222.—Walschaert valve-gear with eccentric.

Another model of this valve-movement, as used on the compound locomotives of the Italian railways, is shown in Fig. 223. The crank-pin arm operates the motion of the slotted link. The valve-rod block and the rod are balanced by a weight on the rock-shaft arm and operated by a lever connected to the third arm. Valve-lead is made by the cross-head arm-link and lever connected to the valve-rod and link-block rod.

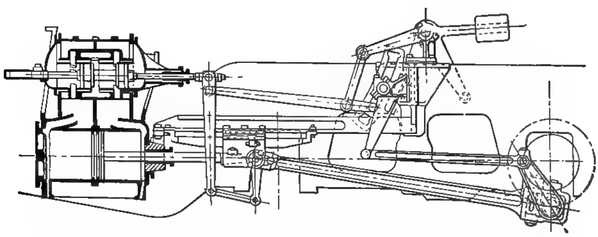


FIG. 223.—Walschaert's locomotive valve-gear crank-pin arm.

A novel reversing-gear without eccentrics is shown in Fig. 224. The valve-stem is connected to the middle of a short link, one end of which are pivoted to the cross-head bar or swinging-lever sliding in an eye or sleeve-block on the cross-head; the opposite end of the

link is pivoted to a radial bar and to the slide on the link-block. This block receives a vertical motion from a sliding-block on the connecting-rod, which is kept in position by a rod pivoted on the cylinder-head. The cross-head swinging-bar imparts a movement to the valve-stem equal to the lap and lead of the valve. The lateral motion of the connecting-rod operates the throw and reversal of the valve by the position of the sliding-link as controlled by a hand-lever.

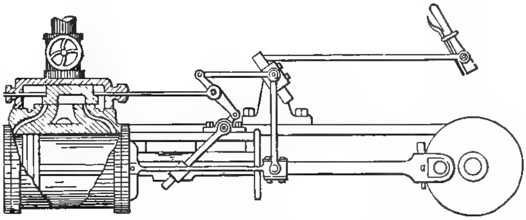


FIG. 224.—Reversing-gear.

A floating valve-gear, used on the reversing-ram of large marine engines, and having peculiar features in its movement, is shown in two positions in Fig. 225. The floating-lever *g* is connected to the cross-head at *k*, and pivoted to the valve-rod at *h* and to the reverse-lever rod at *i*. The piston-valve is indirect, and takes steam at its centre. When the lever *d* is set vertical the action of the valve sends the piston to the centre of the cylinder, and when pushed over sends the piston in the opposite direction. The springs at each end of the traverse-bar are to prevent shock by the sudden movement of the piston.

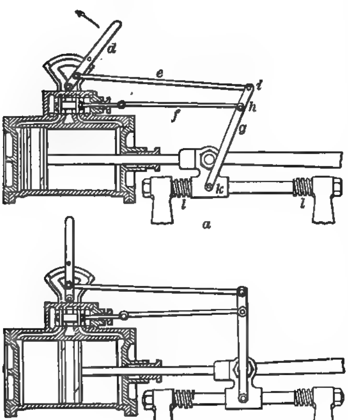


FIG. 225.—Floating valve-gear.

A novel valve-gear has been applied to a three-cylinder engine, illustrated in Fig. 226, in which the piston-valves are operated by a connecting-rod from the valve to the trunk of the following piston. The exhaust is discharged into the main trunk of the engine through the hollow spool-valves, and from the ports opened by the trunk-pistons into jacketed recesses. Steam-connection is made with the chambers at the head of each cylinder.

Another curious engine is the "Brotherhood" three-cylinder

engine, of English origin, in which the steam enters and fills the central chamber with equal pressure on all the pistons. The valve is of the rotary-disk type, operated by the crank-pin within the

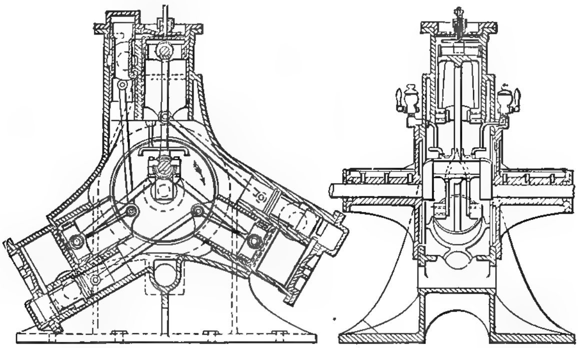


FIG. 226.—Three-cylinder engine.

chamber, and gives steam to the outside of the pistons alternately through an outside port to each cylinder. The steam-passages cover the shaft, making a steam-tight stuffing-box necessary on the shaft. Its advantages are compactness and convenience for small powers, but it lacks efficiency.

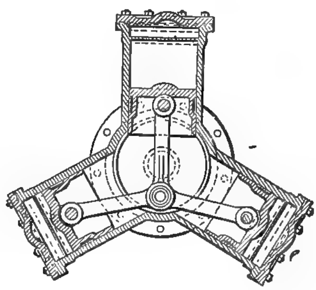


FIG. 227.—"Brotherhood" type of three-cylinder engine.

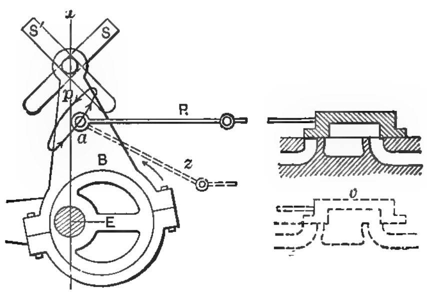


FIG. 228.—Wolf reversing-gear.

In regard to the various types of rotary engines and the hundreds of different models and valve-gear motions which have passed the ordeal of trial and failure, we can only say, from our own experience and knowledge of their temporary life, that the rotary principle, as developed in its long career, has finally found its success only by going

back to the type of the early ages—the simple reaction of the Hero and Avery models—with the addition of the multiple effect, and culminating in the modern steam-turbine.

The reverse-gear of the Wolf model is another form for reversing from a single eccentric, and is shown in Fig. 228.

The end of the eccentric-arm B is pivoted to a block in the slotted link S, which is also shown in its opposite position for reversing at S′; the valve-rod R, being connected by a pivot to the eccentric-arm at *a*, acquires an elliptical motion by the action of the eccentric and the link-block, which becomes vertical or reversed by throwing over the link.

A most novel valve-gear for a triple-expansion engine from a single eccentric is used on the engines of the Edison Electric Company, New York City, and is illustrated in Fig. 229. The eccentric-arm is pivoted by a link-arm to the frame at A, which carries a pin off from its central line, and connects with the high-pressure valve-rod. The bell-crank lever B is pivoted by a link to the lower side of the eccentric-strap, and from its upper arm is pivoted to the mean-pressure cylinder valve-rod; the low-pressure valve-rod is a direct-line connection through a rocker-shaft and arms at C. This is the most ideal conception for operating the valves of a triple-expansion engine yet brought to the notice of the author. It is a study for the curious in valve-gear motion.

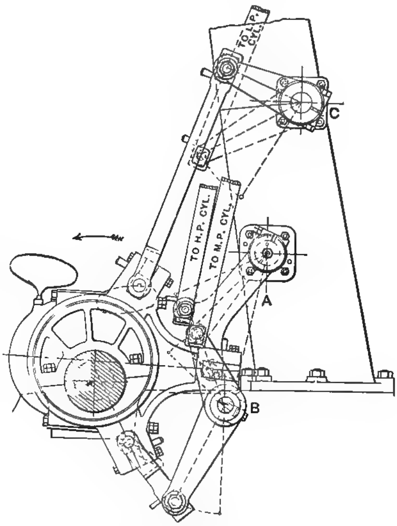


Fig. 229.—Triple-expansion valve-gear.

A valve-gear derived from the elliptical motion of a pin near the middle of the connecting-rod is the ideal of the "Joy" valve-gear movement. The ellipse made by the path of the pin is symmetrical with the central line of motion in the engine; but the action of the link-movement slightly changes the direction of its axis in regard to

the valve-motion. Its application to a vertical engine is shown in Fig. 230, with a screw-adjustment for the position of the link.

In Fig. 231 is shown the same arrangement applied to a horizontal engine, with a connecting-rod, A, extending to a lever for reversal.

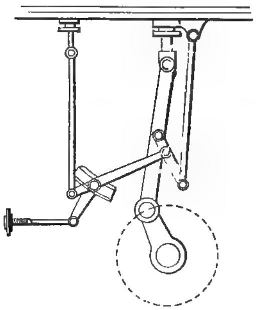


FIG. 230.—Vertical valve-gear.

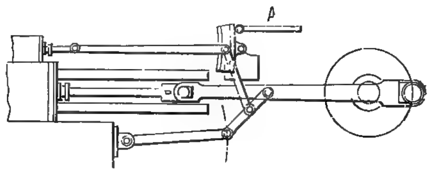


FIG. 231.—Horizontal valve-gear.

In Fig. 232 are given a more defined diagram and description of the Joy valve-gear.

The end of the lever *abc* is guided by the rod *gc*, and is attached to the connecting-rod at a point, *a*, which describes an ellipse, having the length a_1a_2 equal to the stroke of the piston. This ellipse, which is omitted to avoid confusion, is symmetrical with regard to the axis XX′, and is slightly more pointed at the crosshead-end than at the crank-end.

The point *b*, which describes the irregular ellipse bb_1b_2, takes the place of the single eccentric used on other valve-gears, and acts on the lever *bi*, while *e* is guided on the circular arc ff_1 by the sliding-block B, and the point *e*, which describes the ellipse ee_1e_2, carries the valve-rod *ed*. The connecting-rod CD, the valve-rod *ed*, and the rod *cg* are in the same plane; the levers *ac* and *be* and the curved guide-bars ff_1 are double, one set of levers being on each side of the connecting-rod. In the

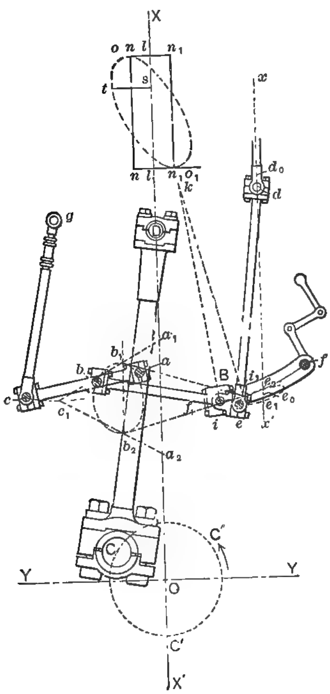


FIG. 232.—Joy valve-gear details.

drawing the system of levers in front of the connecting-rod is omitted to show the construction more clearly. The point i could be guided on the arc ff_1 by a link centred at e. Such a construction is used in marine engines.

The radius of the guiding-link ff_1 is always equal to the length of the valve-rod.

The irregularity due to the angularity of the lever *bie* is compensated for by the action of the lever *ac*, somewhat in the manner that the linkage known as Watts's parallel motion is made to give nearly a straight line of motion.

The guiding-bars ff_1 are hung on trunnions, with the axis at the point i_1, and are connected at f to the reversing-lever. The gear is shown in full-gear position for left-hand rotation. It may give a shorter cut-off if the guiding-bars ff_1 are given less inclination from the horizontal or mid-gear position, and when in mid-gear it will give the valve a motion equal to the lap plus the lead; and when the guiding-bars ff_1 are inclined the other way, the engine will be reversed.

When properly proportioned the Joy gear gives rapid motion to the valve when opening and closing, less compression at short cut-off than does the link-motion, and the cut-off can be made nearly equal for all positions of the gear. Like many other valve-gears, it gives constant lead. The principal defects are the number of parts and joints that are liable to wear loose, and the obstruction that it offers to inspection and to proper care of the crank-pin and cross-head when the engine is running.

In Fig. 233 are shown the plan and vertical section of the Porter-Allen medium to high-speed engines, made by the Southwark Foundry and Machine Company. The valve-gear, which is of the four-port, balanced slide-valve type, is of peculiar interest, not only in the fact of having four slide-valves, but in the way in which the transmission of motion to the valves from a single eccentric is made. The steam- and exhaust-valves are on opposite sides of the cylinder, and are operated from a single eccentric through independent rock-shafts.

The movement of each valve opens or closes double-port passages for steam and exhaust, as shown by the arrows in Fig. 234. By this construction only narrow seats and short valve-strokes are required to give large port-openings. The arrangement of the valve-gear is clearly shown in plan and elevation in Fig. 233. The eccentric E is

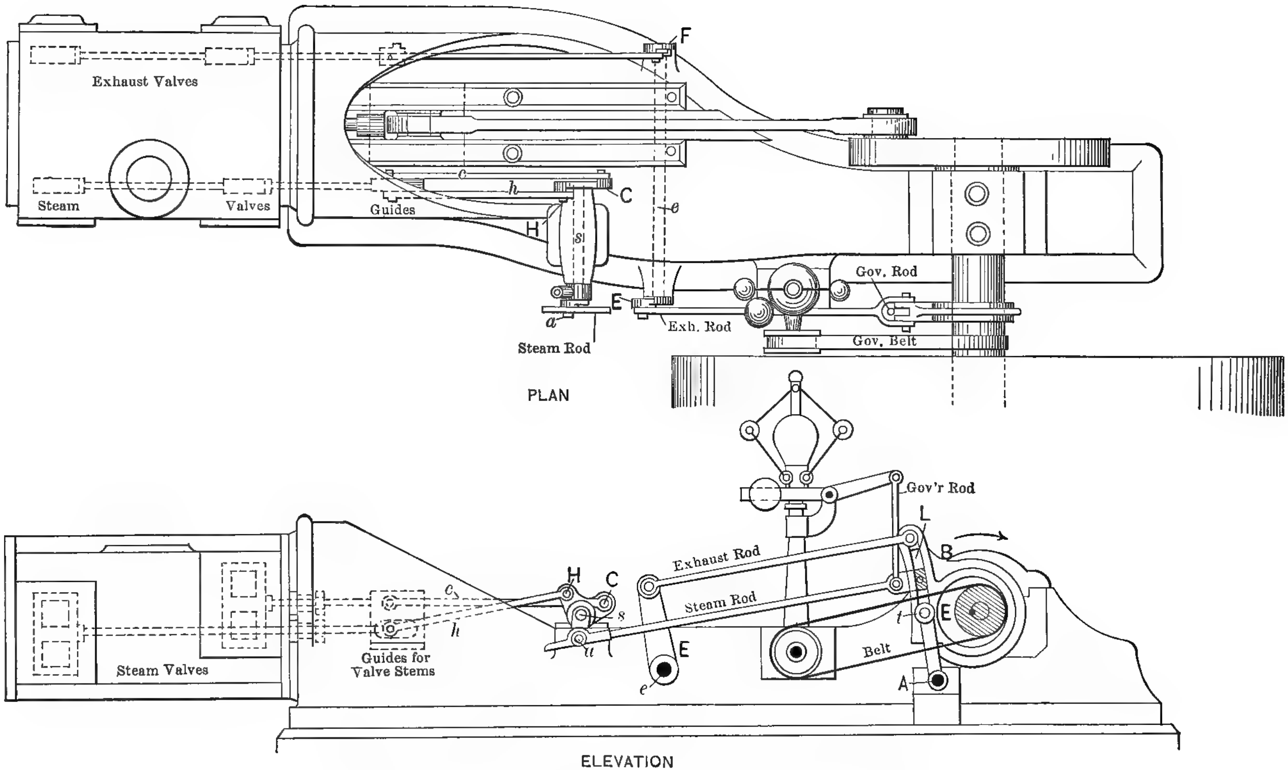


FIG. 233.—Plan and elevation of the Porter-Allen engine.

forged on the shaft and is coincident with the crank. The eccentric-strap and the curved link L are made in one piece, and the link is pivoted at its central point on the trunnions *t*, which in turn are pivoted to the frame at the fixed point A. The vibration or horizontal movement of the trunnions is equal to the throw of the eccentric. In the slot of the link is the block B, from which are driven the two steam-valves. The short rock-shaft *s* on the frame is actuated by the outer arm *a*, which is connected by the steam-rod with the block in the link. It carries on its inner end the two arms H and C, which drive respectively the head-end and crank-end steam-valves, through the medium of the two rods *h* and *c*, and the two valve-stems. The steam-valves are offset in the chest, in order that connection to each valve may be made at its centre; and short guides are provided at the connections of the rods H and C and the valve-stems.

An inspection will show that the link has a peculiar movement, composed of the horizontal and vertical throws of the eccentric. The link is restrained from rising by the trunnions, and the horizontal throw of the eccentric draws off the lap of the valve, while the vertical throw tips the top of the link alternately to and from the cylinder as the eccentric-centre rises or falls in its revolution, the upward throw tipping the link toward the cylinder and the downward throw tipping it from the cylinder. This tipping of the link opens and closes the steam-valves by rocking the rock-shaft by means of the steam-rod and arm *a*.

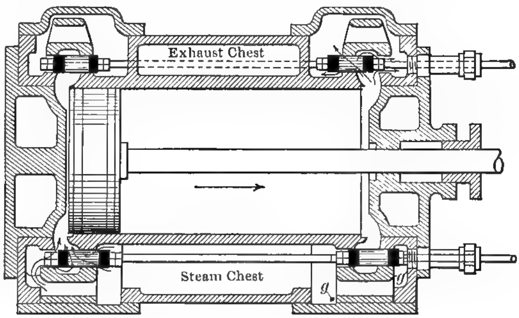


FIG. 234.—Porter-Allen cylinder.

In Fig. 234 is shown a horizontal-plan section of the cylinder and steam-chests, with the double port-opening for both steam and exhaust at the commencement of the forward stroke. The valves are opened and closed quickly by the middle movement of their arms, and have very little movement while open or closed, as the arms are then at the extremes of the travel. The position of the block in the link is under the control of the governor, a dropping

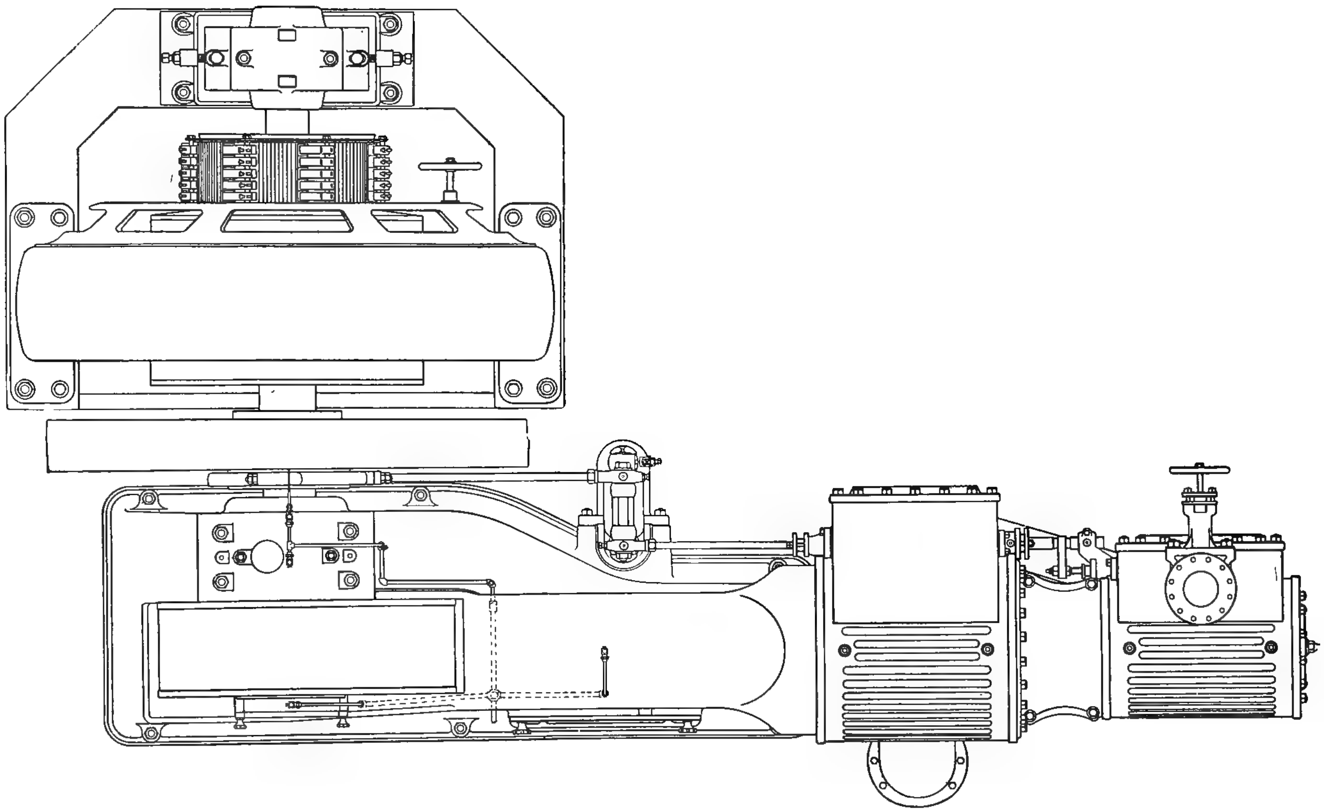


Fig. 235.—High-speed tandem compound engine, manufactured by the Ball Engine Company.

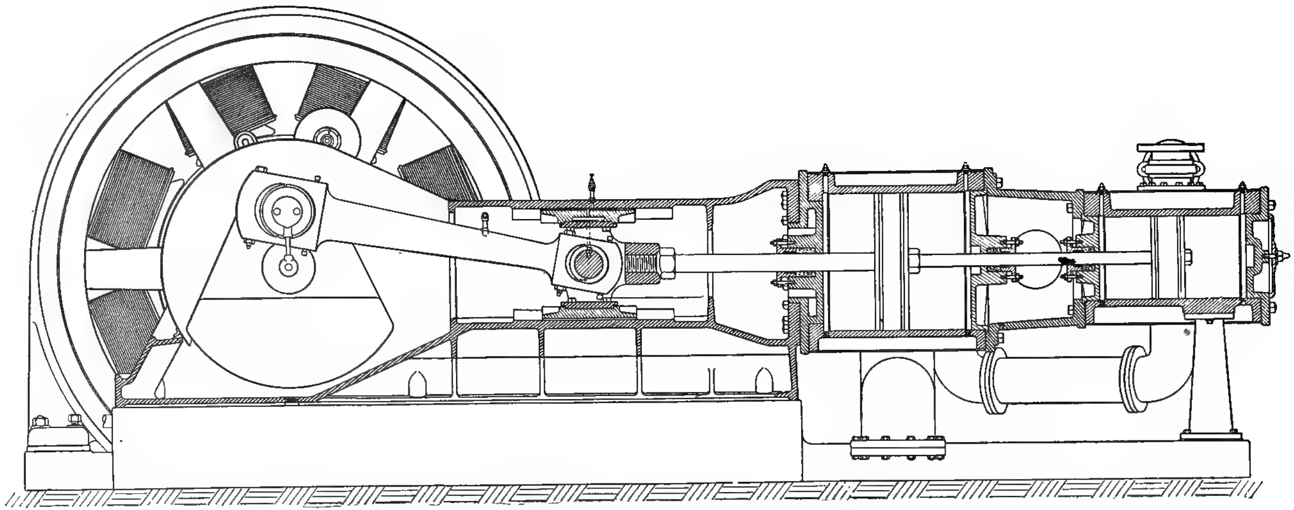


Fig. 236.—Vertical section of the high-speed tandem compound engine, manufactured by the Ball Engine Company.

of speed causing the governor-balls to drop and so raise the block, and an increase of speed forcing the block down toward the trunnions. When the block is at the top of the link, the steam-rod receives the full tipping motion of the link, and cut-off takes place at the maximum point, about six-tenths of the stroke. On the other hand, when the governor-balls are in the extreme upper position the block is forced clear down to the trunnions, and so receives none of the tipping motion of the link. Then the valve is merely opened for lead, and is closed immediately.

Thus the steam-valves are always opened and closed quickly at the mid-travel of their arms; the velocity of cut-off increases as the cut-off is lengthened, since the block is higher in the link, and so corresponds to the increased piston-velocity near mid-stroke; and the velocity of valve-movement is increased directly with the speed of the engine.

The Porter fly-ball governor is used. It is carried on a bracket from the engine-frame and is belted to the crank-shaft. Its distinguishing features are light fly-balls with a high rotative speed to secure sensitiveness, and a heavy ball or weight on the vertical shaft to secure the gravity-effect required to keep the revolving balls in their effective plane.

Figs. 235 and 236 give sketches of the plan and sectional elevation of a high-speed tandem compound engine direct connected to a multipolar generator. The slide-valves of both cylinders are operated from a single eccentric with a rocker-arm and valve-rod extension through the low-pressure steam-chest, on the end of which is an arm attached to the high-pressure valve-rod, with adjustment for its proper operation.

The receiver is simply a pipe-connection from the exhaust of the high-pressure cylinder to the steam-chest of the low-pressure cylinder. The cylinder-volumes are so proportioned that the assigned cut-off in the high-pressure cylinder will equalize the gross pressures in both cylinders for the required speed.

CHAPTER XVI

THE CORLISS ENGINE

In Fig. 237 is shown a full-page view of the Corliss engine, with single eccentric and the usual transmission-gear for operating the valves, and with the names of the various connections, the details of which will be illustrated in the following pages.

The eccentric, by means of the eccentric-rod, rocker-arm, and reach-rod, causes the wrist-plate to oscillate back and forth. On the wrist-plate are placed the steam- and exhaust-pins which operate the steam- and exhaust-links respectively. These links operate arms attached to the valves. The exhaust-valves are attached directly to the exhaust-arms, and they rock back and forth with the wrist-plate, opening and closing the exhaust-ports at the proper time. The steam-valves are operated differently. Each steam-link oscillates a bell-crank, which is loose on the steam-valve stem. On this bell-crank is a latching-gear, which is arranged to take hold of a steam-arm directly attached to the steam-valve. As the bell-crank moves, the steam-valve is thus made to follow it, and thereby open the steam-port.

As the latch on the bell-crank moves, it reaches a stationary knock-off cam, and the further motion of the bell-crank forces the latch against this cam, so that the latch is released and the bell-crank can no longer pull the steam-arm with it. The steam-arm is always acted upon by a downward pull from the dash-pot. Hence, as soon as the knock-off cam causes the latch to release, the steam-valve is pulled shut by the dash-pot and cut-off occurs. The position of the knock-off cam is changed by the governor, so that the time of cut-off varies with the load on the engine, in order to keep the speed constant.

The figure shows the eccentric in the lower central position—that is, with the eccentric vertically downward. Each point on the wrist-plate is then in the centre of its motion. Arrows indicate the directions in which the various parts are moving. The piston has very nearly reached the crank-end of its stroke, and the crank-end steam-valve

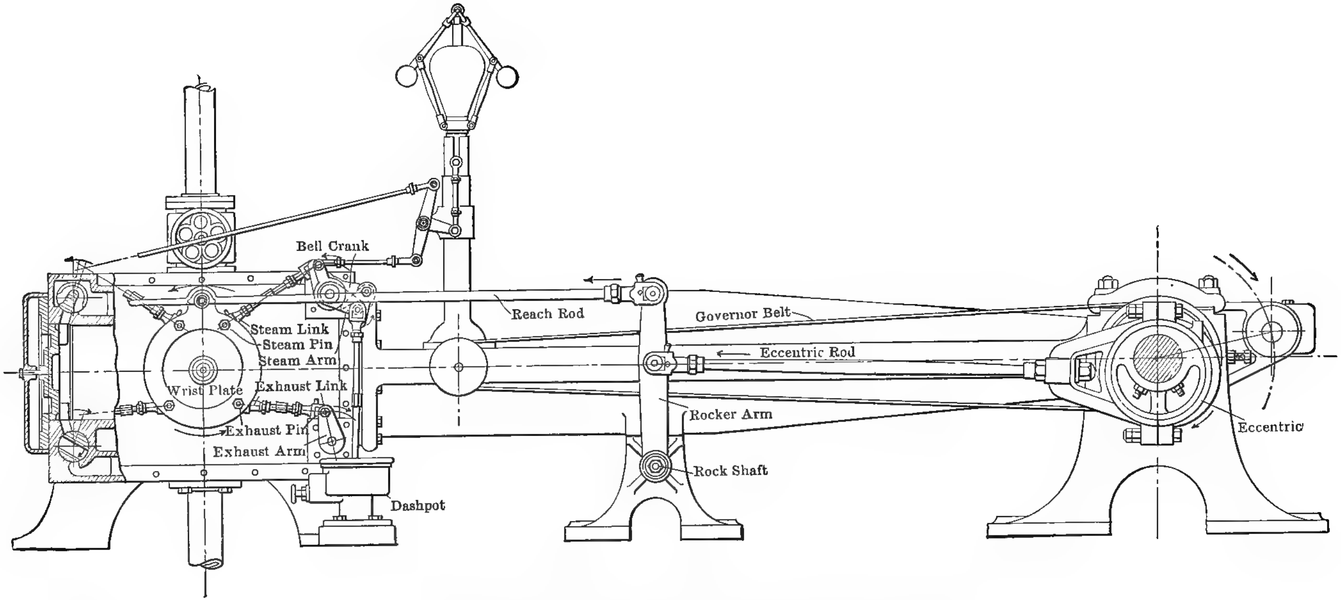


FIG. 237.—General arrangement of the Corliss valve-gear.

is almost ready to open, in order to admit steam to drive the piston on the backward stroke.

The amount by which the crank-end steam-valve closes the port in the position shown is the steam-lap. The head-end steam-valve was pulled shut by the dash-pot some time before the position shown. The bell-crank then moved to the end of its travel by itself, and it is now going back again after the steam-valve in order to pick it up and cause it to open the head-end steam-port at the proper time.

Since the wrist-plate is in the centre of its motion in the position shown, the crank-end exhaust-valve is in the same situation as is shown for the head-end exhaust-valve, but it is moving in the opposite direction with respect to its port, and is therefore just closing after having caused compression. The amount by which the exhaust-valves close the ports in the position shown is called the exhaust-lap.

Each of the valves moves back and forth as the eccentric moves back and forth, exactly as would be the case with the various edges of a common slide-valve. There is a distorting effect due to the obliquity of the links in the Corliss gear, but this merely varies the speed with which the valves move. Therefore admission, release, and compression are effected by the eccentric in very much the same way with a Corliss valve-gear as with a common slide-valve.

In all descriptions of the action of the common slide-valve will be found reasons for the use of lap and of angle of advance. For the same reasons, a Corliss valve has lap and angle of advance. The lap of a slide-valve is the amount by which the valve closes the port when both eccentric and valve are in their central positions. The lap of a Corliss valve is the amount by which the valve closes the port when the eccentric and wrist-plate are in their central positions. However, the valve itself is not then in the centre of its travel, owing to the distorting effect of the motion of the link and wrist-plate.

The eccentric of a Corliss engine is therefore placed ahead of the crank by 90 degrees plus an angle of advance. In Fig. 237 the angle of advance is the angle between the crank and the horizontal centre line. The latest point of cut-off is somewhat less than half-stroke. The less the angle of advance the nearer it is to half-stroke. Hence, in order to increase the capacity of the engine, the angle of advance is made as small as possible. This is done by making the percentages

of compression and release as great as possible, since the angle of advance is determined by a point half-way between the two.

The compression must occur early enough to give a proper cushion. It varies from 90 to 98 per cent., according to circumstances. The release must occur early enough to give a proper exhaust-lead. The exhaust-lead is the amount that the exhaust-valve is open when the piston starts on the exhaust-stroke. If the exhaust-lead is insufficient, the exhaust will be restricted at the beginning of the exhaust-stroke, giving an indicator-diagram with "turned-up toes." On the other hand, a too early release must be avoided; otherwise the end of the expansion-line will be lowered. It will usually be found that if release occurs at from 98 to 99 per cent. of the stroke, the exhaust-lead will be sufficient.

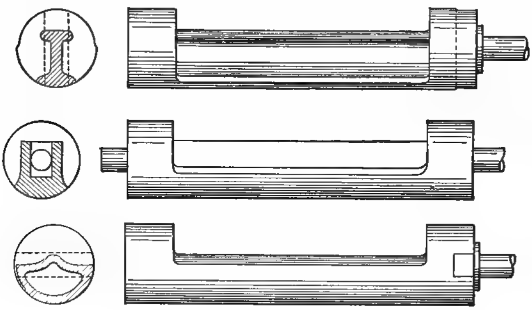


FIG. 238.—Corliss valves.

The details of the valves and valve-gear of the Corliss type are variable to a great degree, and we can only illustrate a few of the leading lines of design. In Fig. 238 are shown two forms of steam-valves and one of the exhaust-valve in general use with a single eccentric. The stem of the uppermost valve is mortised vertically into the valve, which gives the valve a free adjustment for perfect seating. The stem in the middle figure passes entirely through the valve with rectangular bearings, while the stem of the exhaust-valve works in a horizontal mortise.

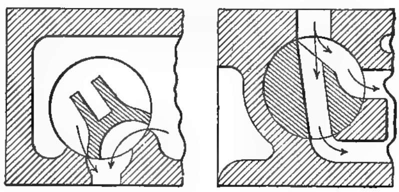


FIG. 239 —Double-ported Corliss valves.

In Fig. 239 are shown sections of the double-ported steam- and exhaust-valves and their action, as shown by the arrows. This model of valve is operated by double eccentrics and double wrist-plates, which allows a greater range of cut-off than practicable with a single eccentric and single wrist-plate.

The single eccentric and single wrist-plate allow of the proper opening and closing of the steam- and exhaust-ports with regard to cut-off, exhaust, and compression for a cut-off not later than one-half to five-eighths of the stroke, while the double eccentric and double wrist-plate give a possible cut-off at nearly full stroke in case of overload on the engine.

In Fig. 240 is shown a single-eccentric valve-gear with overpull steam- and exhaust-links, as well as the right- and left-threaded

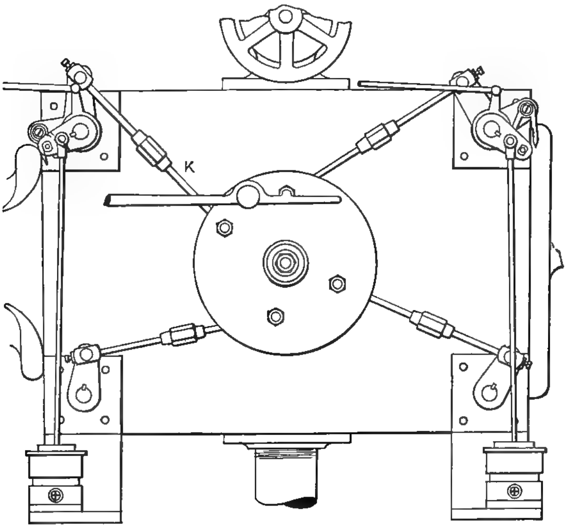


FIG. 240.—Single-eccentric valve-gear.

adjustment-couplings and lock-nuts, and in Fig. 241 a double eccentric valve-gear on a single centre with overpull steam- and exhaust-links.

The matter of arranging the links and bell-crank movements as to overpull or underpull of the links and the bell-crank action depends much upon the opinion of designers, in regard to the velocity-stroke of the valve, as to which side of the valve may be considered best for steam-inlet and exhaust. Four or more bell-crank valve-lever positions for inlet and exhaust are in use by the leading steam-engine builders in the United States.

Fig. 242 shows the arrangement of the single wrist-plate and its link-rod connections, with the valve-levers of the steam side turned

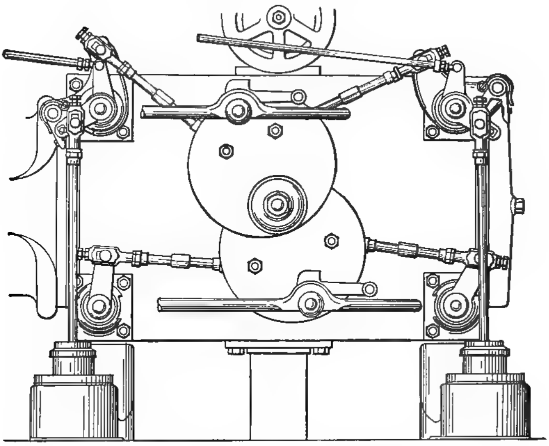


Fig. 241.—Double-eccentric valve-gear.

downward and those of the exhaust turned upward. The position of the wrist-plate and valve-motion is in the middle of their travel.

In Fig. 243 are shown an elevation and plan of the valve-motion, to which is attached Cité's releasing-valve gear. A is the valve-stem

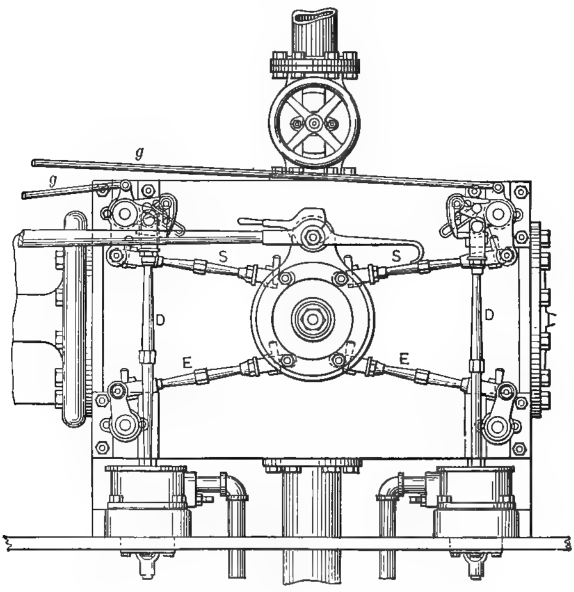


Fig. 242.—Fishkill-Corliss valve-motion.

and B the valve-lever, and CC′ a bell-crank which vibrates loosely on a sleeve around the valve-stem, and is connected by an adjustable link-rod to the wrist-plate. The end of the arm C carries a small rock-shaft, D, which has a hook, E, fastened on one end. This hook is provided with a hardened steel catch-plate, *b*, which engages a similar plate, *c*, fastened on the end of the valve-lever B, and the hook is kept in place by a light spring, *f*. On the end of the rock-shaft D, opposite the hook E, is fixed a forked crank F, having a pin on which is mounted a sliding-block fitted to move in a slot, *i*, of a link, G. The link is mounted at and vibrates about a point, *j*, in one arm of a bell-crank, H, and the bell-crank oscillates upon a sleeve around the valve-stem. The other arm of the bell-crank H is connected by an adjustable rod, Z, to the governor. By an arrangement not shown, if the action of the governor become deranged by the breaking of the belt, the sudden dropping of the governor-balls below their ordinary limit for speed reverses the releasing-gear, and the block in the slide *i* is pushed out and prevents the hook E from catching the valve-lever. In the ordinary regulation for speed the block will have been pushed so far outward that it will have slightly turned the small rock-shaft D, and moved the hook E enough to release the valve-lever B. Then the dash-pot will act and close the valve. At this moment of release, effected by the toggle-like action of the link, the pressure on the bell-crank H, caused by the liberation, will be exerted in a radial line from the centre of the slot through the point *j* to the centre of the valve-stem or the stand which supports it, and during the entire movement of the hook E there will be no appreciable strain to turn the bell-crank H, and consequently there will be no strain to disturb the normal action of the governor. As the position of the bell-crank H is controlled by the governor, any

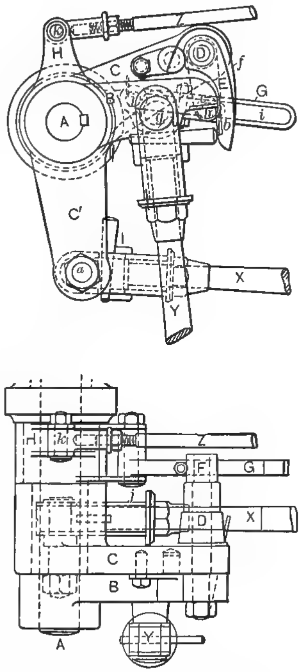


FIG. 243.—Valve-gear of the Fishkill-Corliss engines

change in the height of the governor will cause a change in the position of the point j, and a corresponding change in the time of release.

In the following drawings are illustrated the conditions and limitations of the valve-action in the single-eccentric Corliss engine.

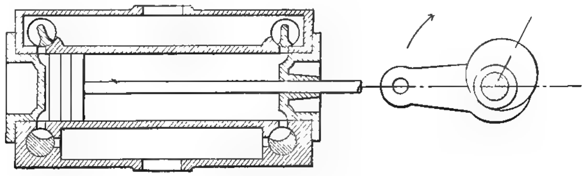


Fig. 244.—Angle of advance.

Fig. 244 shows the eccentric set at its angle of advance and the steam-valves adjusted for the lead, with the crank-pin on the inner dead-centre, the closed ports having sufficient lap to insure a steam-tight joint.

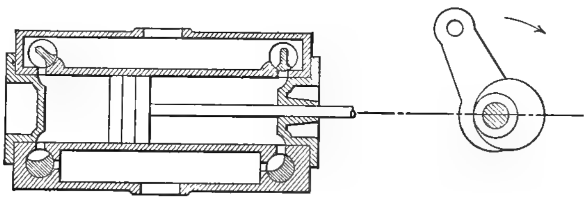


Fig. 245.—Point of cut-off.

The next phase is the point of cut-off, shown in Fig. 245, in which the steam-valve of the forward stroke is to suddenly close from a full opening at one-third stroke by the release-gear and by the pull of the dash-pot with the exhaust-valve at full opening.

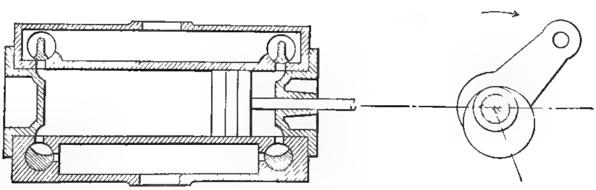


Fig. 246.—Commencement of compression.

The last phase is the commencement of compression, illustrated in Fig. 246, and represented at about one-sixth of the stroke, with the

forward exhaust-valve just closing and the other valves entirely closed with their proper laps, due to the required adjustment of steam- and exhaust-valve links.

In the next three illustrations (Figs. 247–249) the angle of advance of the eccentric is set at a less angle, say about 100 degrees, from the inner dead-centre for one-half cut-off.

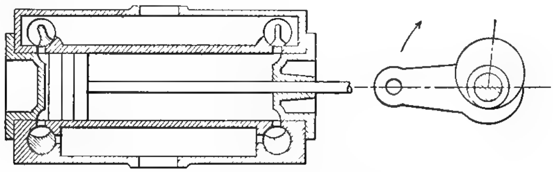


FIG. 247.—Angle of advance.

In Fig. 247 the eccentric is set at about 10 degrees ahead of a right angle from the crank-pin for one-half cut-off, showing the steam-valve open by the amount of its lead and the exhaust-valve just opening.

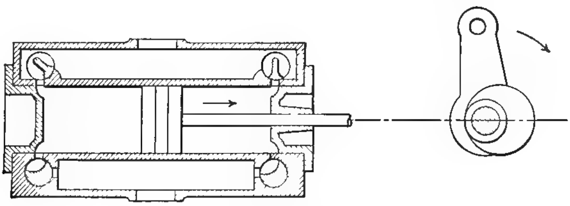


FIG. 248.—Point of cut-off one-half.

In Fig. 248 the point of cut-off is advanced to one-half the stroke, which takes place at the extreme throw of the eccentric, which, with the required adjustment of the links, gives a release of the steam-valve from full opening and with a full opening to the exhaust-valve.

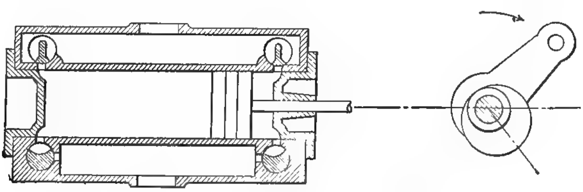


FIG. 249.—Commencement of compression

Fig. 249 shows compression slightly less than in the first set of illustrations, Figs. 244–246, and near the extent of the action of a single eccentric for the best efficiency in steam-distribution. The advance of the eccentric may be lessened to 90 degrees from the crank-pin, or given a negative place with advance of the cut-off to five-eighths, but with restricted port-openings, which can only be given their best conditions by double-eccentrics for extended cut-off to meet overload.

For the benefit of students and others interested in the numerous designs of releasing-gear for Corliss engines, we give the following illustrations and descriptions.

A standard model of release-gear, illustrated in Fig. 250, is that used by the Fishkill Machine Company and others, in which, A being the valve-stem, a bell-crank operated by a connecting-rod from the

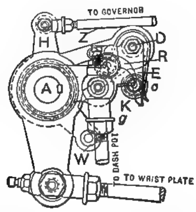


FIG. 250.—Bell-crank knock-off.

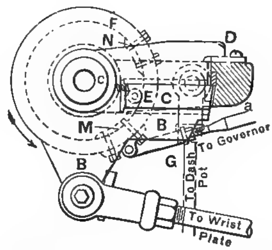


FIG. 251.—Bass release-gear.

wrist-plate lifts the grab-hook E and the valve-arm. An adjustable roller at R releases the valve-arm, which is pivoted to the dash-pot rod for regulating its fall. The release-roller R is operated by the bell-crank H and rod Z from the governor.

The release-mechanism used on the Bass engines, built by the Bass Foundry and Machine Company, is shown in Fig. 251, in which the grab-hook consists of a block, C, sliding in a grooved slot in the bell-crank lever B, B, and normally forced out to catch the block on the rocker-arm at D by a spring. The block C carries a pin, E, on the rear side, which is held in contact with a cam-ring, F, having two knock-off dies, M and N, on its inside surface. As the bell-crank moves in the direction of the arrow from the position shown, the roller on the pin E strikes the cam-die N and is forced rapidly inward, releasing the drop-block a.

The release-gear used on the Allis-Chalmers engines is shown in Fig. 252. The hook H, which is forced inward by the spring, engages with the valve-lever B, and as the bell-crank lever A, A moves in the direction of the arrow, the valve-lever B is lifted and opens the valve, and at the proper moment, as regulated by the governor, the trip-lever T comes in contact with the projection N of the cam C, forcing it and the grab-hook outward and releasing the drop-lever B, which is brought down by the action of the dash-pot.

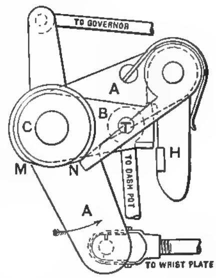


Fig. 252. — Allis-Chalmers release-gear.

In Fig. 253 are shown two positions of the release-gear used on the engines of the Filer & Stowell Co.

In this design B is the bell-crank, which carries the hook H, mounted on a short shaft, and on the other end of which is the trip-lever (not shown), which engages with the knock-off cam C, operated by the governor-rod. K is the drop-lever with dash-pot connection. The cam-lever C, controlled by the governor, limits the time of release by the hook H. The lower figure shows the position of the parts at the moment of release.

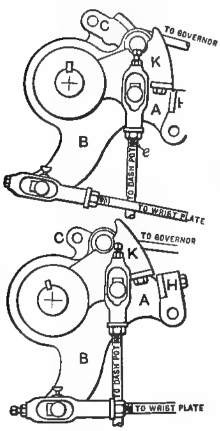


Fig. 253.—Trip valve-gear.

The valve-gear used on the engines of the Nordberg Manufacturing Company is shown in two positions in Fig. 254, similar in principle to those in Fig. 253, but with an entire change in the position of the operating parts. The curved bell-crank B carries the grab-hook D, mounted on a short shaft, and having an arm at the other end with a trip-lever, *d*, which rides on the knock-off cam A, the position of which is controlled by the governor by pushing the knock-off cam under the grab-hook lever for release.

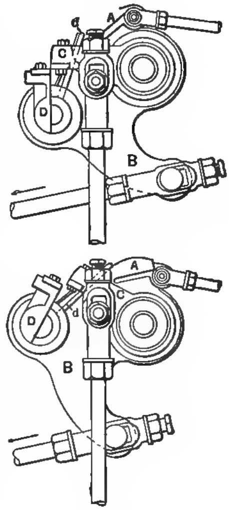


Fig. 254. — Nordberg valve-gear.

The lower figure shows the position of the parts at the moment of release.

The valve-gear on the Sioux City and other engines is shown in two positions in Fig. 255. It consists of an inverted or overhead wrist-plate connection of the bell-crank lever with a forked grab-hook, in which A, A is the bell-crank, on one arm of which is pivoted the forked grab-hook H, held by the spring S, the other end of the fork riding against the cam C, the movement of which by the governor releases the valve-arm B. The lower figure shows the position of the parts at the moment of release.

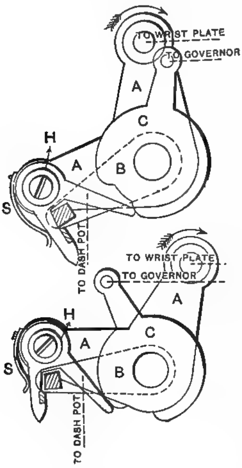


FIG. 255.—Sioux City valve-gear.

The release-gear of the Scottdale Foundry and Machine Company is shown in front and side elevations in Fig. 256. The opposite arm of the bell-crank A carries the latch-block B, which in moving forward engages the block C on the valve-arm D, to which is also attached the dash-pot rod. The latch-block is pressed down by a spring and adjusting thumb-screw at B, and is disengaged by the cam E acting upon the lever G on a rock-shaft in the valve-arm. A cam at J, also on the governor-arm, is a safety-cam, and lifts the lever G and prevents engagement of the latch-block, if the governor-balls should fall by the breaking of its belt.

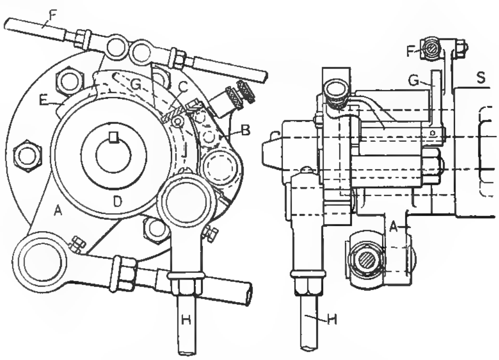


FIG. 256.—Scottdale releasing-gear.

A simple and effective releasing-gear, consisting of few parts, is shown in Fig. 257. It is used on the engines of the Watts-Campbell Company. The action of this gear is well shown in the cut, in which A represents the crank-arm, which is keyed on the valve-stem and carries a steel catch-block a, which is fitted into the end of the crank-

arm or drop-lever. B is the bell-crank lever, one arm of which is connected with the wrist-plate by the rod *b*. In the other arm is fixed the pin C, which carries the rocker-arm D. A small roller *e* is carried at the upper end of the rocker-arm D, and is held against the knock-off cam E by the spring *c*, as shown.

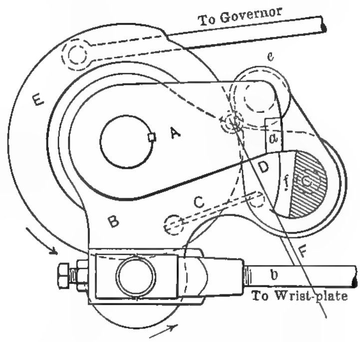


FIG. 257.—Watts-Campbell releasing-gear.

When the bell-crank moves in the direction of the arrow, the edge of the die-block *f* engages the end of the valve-arm A, and raises it to the point of release. At this point the roller *e* at the upper end of the rocker-arm D comes in contact with the projection of the knock-off cam, which forces the roller and upper end of the rocker-arm outward, releasing the arm A, which is rapidly drawn downward by the dash-pot and the rod F.

There are many other models of releasing-gear in use, all of which involve the foregoing principles; but enough have been illustrated for the understanding of their principles of action to meet the ordinary wants of engineers in engineering possibilities, such understanding fulfilling our principal aim.

GOVERNORS AND DASH-POTS

The types of governor best suited for the speed-regulation of engines of the Corliss type appear to be those of the fly-ball and gravity-weight combination, although other models are in use which seem to give satisfactory control.

In Fig. 258 is shown the leading model of the class of the fly-ball gravity-governors, the Porter-Allen, as made by the Southwark Foundry and Machine Company. It consists of a pear-shaped weight, A, moving freely on the spindle, the balls B, B being attached to the spindle cross-bar and to the cross-bar of the weight by rods with forked toggle-joints. A lever, D, pivoted to an arm on the governor-frame and traversed by a pin-connection with a ring in the grooved sleeve

at the bottom of the gravity-weight, is connected to the valve-gear at E and adjusted by a counterweight at C.

A modification of this governor as made by the Watertown Engine Company is shown in Fig. 259, in which a much larger movement of the lever F and knock-off cam-rods E, E is obtained. The upper yoke, to which the arms of the governor-balls are pivoted, is fitted loosely to the spindle, the latter carrying a rack with which mesh the extensions of the arms to which the balls are attached, these being in the form of a sector of a gear. When the balls fly outward the ful-

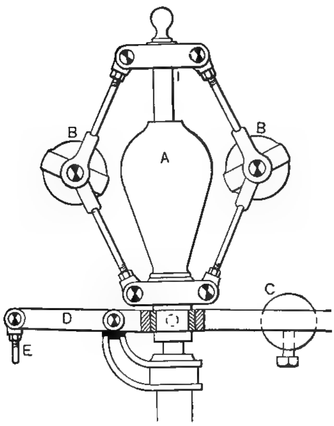


FIG. 258.—Porter-Allen governor.

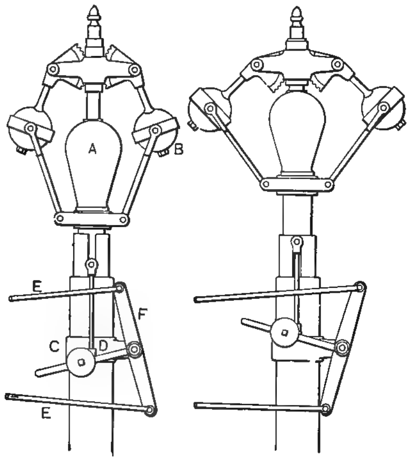


FIG. 259.—Watertown governor.

crum of the arms carrying the balls is raised a certain distance, which increases the height of the central weight over that due to the elevation of the balls, so that a slight change in the position of the balls will cause the weight, and consequently the knock-off cams, to move a much greater distance than that due to the movement of the balls only. The twofold action thus obtained makes an exceptionally sensitive governor, without excessive travel, and without a jerk or a tendency to fluctuate.

The governor used on the Lane & Bodley engine differs materially in its generating action from the models last described and is illustrated in Fig. 260. The governor-balls are fixed at the ends of bell-crank levers which are supported by a circular plate. The horizontal arms

of the bell-crank levers are provided with rollers for the purpose of reducing the friction between them and the collar and sleeve which surround the spindle.

The outward and downward movement of the balls is resisted by the spring which opposes the upward movement of the sleeve. At the lower end of the sleeve is a groove in which is fitted a collar carrying one end of the bell-crank lever, to which is attached the rod operating the knock-off cams, the cam-connections being over and under with a continuous rod. It will be noticed that a very slight vertical movement of the collar will move the lower end of the lever and the governor-rods a considerable distance. The speed of this governor is about 200 revolutions per minute, which renders it capable of producing a great change of centrifugal force for a slight variation in speed and with a corresponding insensibility to varying internal resistances. While the changes in position are made very quickly, there is no jerk or vibration, for the dash-pot at the base of the governor-post prevents a very sudden upward or downward movement of the spindle. The action of the automatic stop S may be readiy understood, and is shown in its working position.

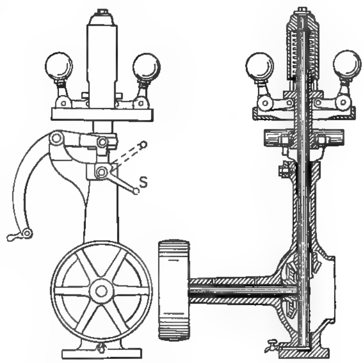


FIG. 260.—Lane & Bodley governor.

When the engine is to be started, the handle on the stop is raised, which raises the governor-sleeve by means of a cam, and moves the knock-off cams on the valve-stems enough to allow the hook to engage with the catch-hook.

As soon as the governor comes up to speed, the governor-sleeve rises, which allows the weight of the stop-handle to turn the cam and bring the lower face under the sleeve, so that, should the governor stop, the sleeve will descend low enough to allow the knock-off cam to prevent the hook from engaging with the catch-block, thus insuring against the opening of the valves.

In Fig. 261 is represented the slow-speed governor used on the Scottdale Corliss engine. Its lifting-power is augmented by the lever-connections on long arms and light balls, which, at a low velocity,

about 60 revolutions per minute, give sufficient lifting-power to operate the gravity-weight. The bell-crank is connected to the sliding-sleeve by a short rod, and upon the other end of its shaft are fixed a lever and its adjusting weight. A retarding annular dash-pot around the spindle regulates the action of the balls and also catches the oil dripping from the collars. The governor is also provided with an automatic safety-stop, supported, when not in action, by the governor-belt, and requiring no attention when the engine is started or stopped.

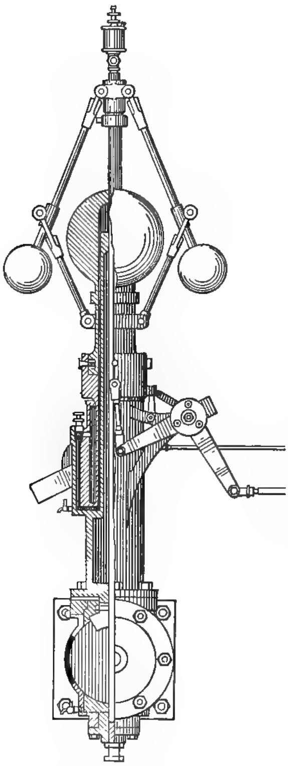


FIG. 261.—Scottdale governor.

Of the great variety of pulley- or fly-wheel governors there seem to be two classes as regards manner of movement of the eccentric, one of which consists in shifting the eccentric and the other in rotating the eccentric on a fixed centre. The mechanism for obtaining these movements is mostly of the centrifugal order, but inertia and positive mechanical devices are also in use.

Of the shifting eccentrics, we illustrate in Fig. 262 the simple device used on the Sweet straight-line engine. In this governor the slotted eccentric has two opposite arms, one end of which is pivoted to an arm of the pulley or fly-wheel, and the other to a link, moved by a lever and by the centrifugal action of a weight, and restrained by a leaf-spring.

In Fig. 263 is shown another design of shifting eccentric. In this the two weights are connected for equal movement by a link, and each balanced by a helical spring. The eccentric-arm is pivoted to an arm of the pulley or fly-wheel and also to the end of one of the weights, so that excess of speed throws the eccentric toward its centre of rotation.

In Fig. 264 is shown the pulley-governor used on the engines of the Fitchburg Steam Engine Company. It will be seen that when the weights *c*, *c*, respectively, are moved outward by centrifugal force,

they draw the eccentric across the shaft by means of the links *d, d*. The weights *e, e* are for the purpose of counterbalancing the weight of the eccentric-strap and one-half the weight of the eccentric-rod in horizontal engines, and the total weight of these members in vertical

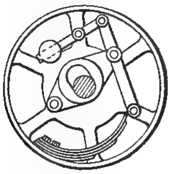


FIG. 262.—Sweet's governor.

FIG. 263.—Shifting eccentric-governor.

engines, which leaves no work upon the governor-weights but to shift the eccentric when the load upon the engine changes. The springs oppose the outward movement of the governor-weights *c, c*, by means of which the speed of the engine may be changed.

It will be apparent that by tightening the springs the speed of the engine may be increased, and by loosening them the opposite result will be obtained. The maximum range of cut-off is from zero to three-fourths stroke, which enables the engine to maintain a uniform speed under wide variations of load.

Of the rotating eccentric-governors there are a number of designs, a few of which we select to show the main points of variation.

In Fig. 265 is shown the pulley-governor used on the Buckeye engine,

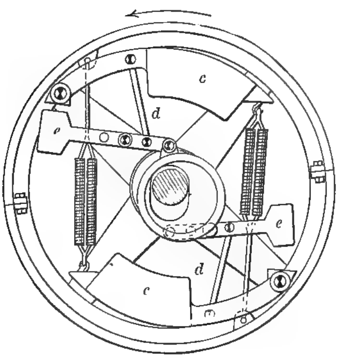


FIG. 264.—Fitchburg pulley-governor.

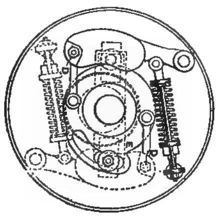


FIG. 265.—Rotating eccentric-governor.

in which the centrifugal force of two pivoted weights connected to a spiral-slotted face-plate by the links D, D oscillates the face-plate. An arm, E, on the eccentric carries an adjustable wrist-pin by which

the eccentric is rotated, and on an extension of the arm on the other side of the eccentric is a wrist-pin which allows of a reversion of the eccentric.

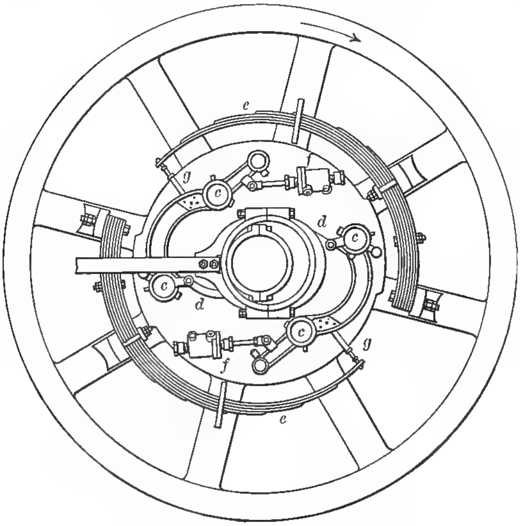


FIG. 266.—Dash-pot governor.

In Fig. 266 is shown the governor of the McIntosh & Seymour Co. on their multiported four slide-valve engines. In this governor the cut-off eccentric is free to revolve on the shaft, and is connected to the weights *c*, *c* by the links *d*, *d*. The outward movement of the weights is resisted by the leaf-springs *e*, *e*, while any tendency to sudden fluctuations in the position of the weights is prevented by the dash-pots *f*, *f*. The ends of the springs are connected to the weights by means of telescopic pins, *g*, *g*; lengthening these pins increases the compressive effect of the spring, and thus offers greater resistance to the weights. To increase the sensitiveness of the governor, therefore, these pins must be lengthened, and if the governor is too sensitive they must be shortened. The circular openings in the weights are provided with extra weights in the form of bushings. The speed of the engine is changed by changing these bushings, inserting heavier ones when the speed is to be reduced and lighter ones when it is to be increased.

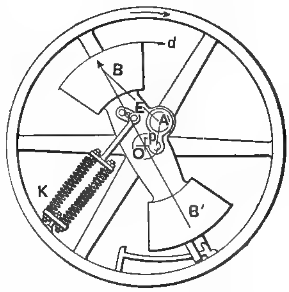


FIG. 267.—Inertia-governor.

In Fig. 267 is shown an inertia-governor, in some designs of which the momentum of the weights and the centrifugal force are combined factors in the operation of varying the position of the eccentric, either by rotation or by shifting it from its centre. The illustration shows

the inertia-governor used on the Leffel engine. The weights B, B′ are balanced, with their centre of gravity at the centre of the engine-shaft, but pivoted at A to the fly-wheel, and by an arm to the eccentric at p. The spring K holds the weights to their normal position, their range of motion by differential momentum from variable speed of the engine being limited by the stops on the rim of the fly-wheel or pulley.

THE VACUUM DASH-POT

The dash-pot is one of the most essential adjuncts of the Corliss type of engine. The speed and softness of the valve-closure are due to the perfect action of the dash-pot, and as there are several designs in use we illustrate some of their features and manner of action.

In Fig. 268 is shown a section of the dash-pot used on the Frick engine. It is a dustless dash-pot, as the air that is drawn under the piston in its upstroke is exhausted into the same annular chamber from which it is taken, which also renders the dash-pot noiseless in operation. As the plunger P is drawn upward by the valve-gear, air is drawn into the plunger-cylinder from the annular chamber A, through the check-valve c. The air is not sufficient, however, to prevent the formation of a partial vacuum, which draws the plunger quickly downward when the valve-spindle is released. As the plunger nears the bottom of the cylinder it is cushioned by the air, which is forced back into the outer chamber through the poppet-valve V. The degree of cushioning can be accurately adjusted by means of the screw S.

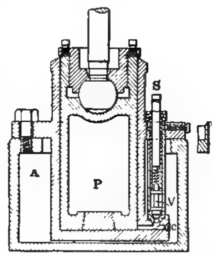


FIG. 268.—Frick dash-pot.

In Fig. 269 is shown in section and elevation the dash-pot used on the Watts-Campbell engines. This dash-pot is very simple, and consists of a cup-cylinder having a tapered lower portion which surrounds the plunger, as shown. The plunger is made to fit over the central column in the cylinder, so that when the plunger is drawn upward a partial vacuum is formed in the space between the plunger and the column. The annular space around the piston-end of the plunger allows a free fall of the plunger by the vacuum effect until

it reaches the taper closure near the bottom of the cup-cylinder. The cushioning effect is produced by the escape of the air from the annular space at the bottom of the cylinder, which continues until the plunger reaches the straight portion of the bore.

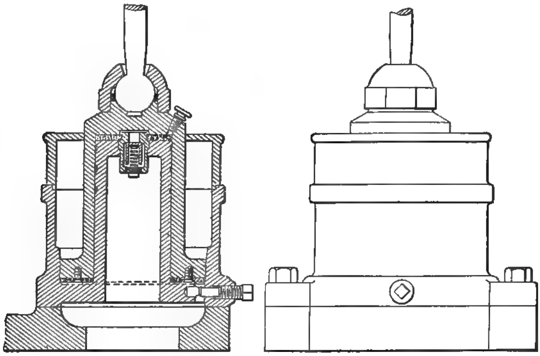


FIG. 269.—Cup-cylinder dash-pot.

The remainder of the air thus entrapped is expelled through the small valve, which offers greater or less obstruction to the small exhaust-port shown at the bottom. A spring check-valve in the head of the central column is an air-relief, and the small screw at the top regulates the vacuum.

SETTING THE VALVES AND GEAR OF A CORLISS ENGINE

Every engineer on taking charge of a Corliss type of engine should have at hand a detailed description of its special action and government, with directions of the builders for setting and adjusting the valves and valve-gear. The variety of motions and kinks in regard to the setting and proper adjustment of valves and gear, including rods, links, eccentric, governor, and dash-pots, differs so much in details of design—although the general principles of operation are essentially alike—that only a general description and illustration for setting the valves and valve-gear can be attempted here. It is hoped that they may be useful as an aid to the new engineer when special instructions are not at hand.

With the first examination of the engine to find if every part of

its running-gear is in working order, see that the builders' marks on the valves and cylinder, as also those on the wrist-plate hub, are in alignment when the wrist-plate is on its central position and the centre line of the rocker-arm is in a vertical position, as shown in Fig. 270, with the eccentric at right angles to the central line of the engine and the crank-pin following the right angle plus the lead. The reach-rod and eccentric-rod being both adjustable, any variation from the proper position may be readily made.

In this position, on removal of the back bonnets the builders' marks on the end of the valve and cylinder will indicate the opening

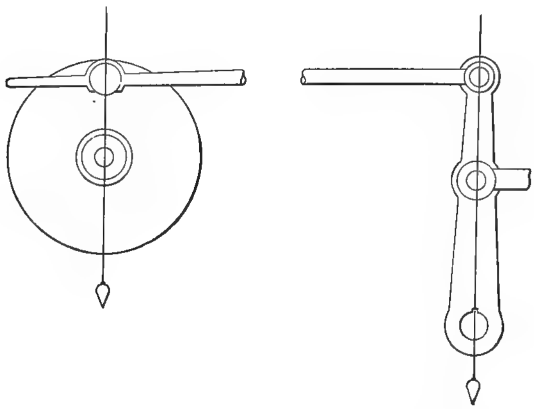


Fig. 270.—Central position of the wrist-plate and rocker-arm.

edge of the valve and port and the lap. On the shoulders of the wrist-plate and its pin find, or make, a mark like *b a* in Fig. 271, which should coincide with the central position of the wrist-plate.

Set the wrist-plate in its central position by an extra washer under the nut of the wrist-plate pin, and then adjust the valves for their proper positions by regulating the link-length for each valve. Then by turning the wrist-plate to its extreme positions, as shown in Fig. 272, connect the dash-pots so that when down the hook will catch with sufficient room to insure locking.

The governor should be blocked up to about the running position, to allow of the free action of the release at this position of the governor-cam.

The wrist-plate should be turned from one extreme position to

the other, so as to open the valves alternately and allow the dash-pot plungers to seat properly, and in order so that the hooks may engage the catch-blocks without fail when the wrist-plate is rocked to its extreme positions. Hook the reach-rod onto the wrist-plate with the eccentric nearly in its extreme position, and adjust its length so that the lines on the wrist-plate hub will nearly coincide. Adjust the length of the governor-rod corresponding to the valve—which is now open—so that the inner member of the hook just engages the projection on the knock-off cam. Move the wrist-plate, by turning the eccentric,

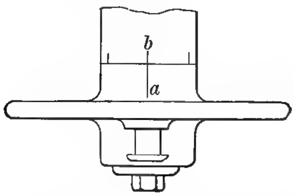


FIG. 271.—Wrist-plate in its central position on the centre.

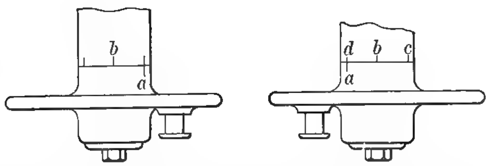


FIG. 272.—Extreme positions of the wrist-plate.

until the lines on the hub coincide. The valve, which has been raised, should now be released and the dash-pot plunger properly seated. The governor-rod should be so adjusted that the steam-valve may be released by the time the wrist-plate reaches the extreme position, in order to insure the valve being closed when the latest point of cut-off is reached, which point corresponds to the extreme position of the wrist-plate and eccentric, and to the governor in its lowest operative position, viz., with the collar or sleeve resting on the safety-stop. The governor-rod at the opposite end of the cylinder should be similarly adjusted.

As the valve-ports open on the inside or outside in different engines, the positions of the builders' marks on the valves must be considered by a reversal of the positions of their connections.

Having made the adjustment of the steam-valves, treat the exhaust-valves in the same way, with the exception of the amount of lap, which should be negative.

The following table shows the usual practice for lap, lead, and exhaust-release for various sizes of Corliss engines, which may be variable to suit the methods of different designers:

TABLE XXXIV.—LAP, LEAD, AND EXHAUST-RELEASE.

Size of engine.	Revolutions per minute.	Steam-lap, inch.	Steam-lead, inch.	Exhaust-release, inch.
12×36–48	90–85	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{1}{32}$
14×36–48	85–75	$\frac{1}{4}$	$\frac{1}{32}$	$\frac{1}{32}$
16×32–48	85–75	$\frac{5}{16}$	$\frac{1}{32}$	$\frac{1}{32}$
18×36–48	80–75	$\frac{3}{8}$	$\frac{1}{64}$	$\frac{1}{16}$
20×42–60	75–65	$\frac{3}{8}$	$\frac{1}{64}$	$\frac{1}{16}$
22×42–60	75–65	$\frac{13}{32}$	$\frac{1}{64}$	$\frac{1}{16}$
24×42–60	75–65	$\frac{7}{16}$	$\frac{1}{64}$	$\frac{3}{32}$
26×48–60	70–65	$\frac{7}{16}$	$\frac{1}{64}$	$\frac{3}{32}$
28×48–72	65–55	$\frac{15}{32}$	$\frac{1}{64}$	$\frac{3}{32}$
30×48–72	65–55	$\frac{15}{32}$	$\frac{1}{64}$	$\frac{1}{8}$
32×48–72	65–55	$\frac{1}{2}$	$\frac{1}{64}$	$\frac{1}{8}$
34×48–72	65–55	$\frac{1}{2}$	$\frac{1}{64}$	$\frac{1}{8}$
36×48–72	62–55	$\frac{17}{32}$	$\frac{1}{64}$	$\frac{1}{8}$
38×60–72	60–55	$\frac{17}{32}$	$\frac{1}{32}$	$\frac{3}{16}$
40×48–84	60–55	$\frac{9}{16}$	$\frac{1}{32}$	$\frac{3}{16}$
42×48–72	70–55	$\frac{9}{16}$	$\frac{1}{32}$	$\frac{3}{16}$

In Fig. 273 is shown a sketch of a Corliss engine of the Hamilton design, with single wrist-plate and trip valve-gear, governor, and dash-pots. The lap-movement of the steam-valves is outside or toward the ends of the cylinder. The exhaust-valves open on the inside

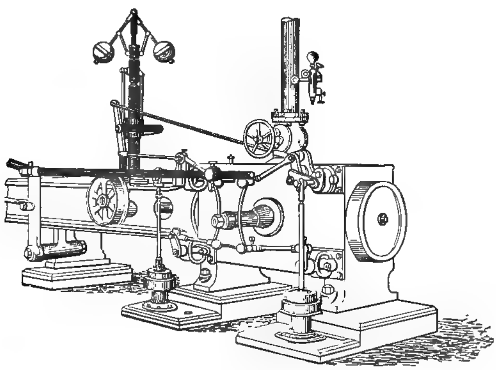


FIG. 273.—Hamilton Corliss engine.

or toward the centre of the cylinder. A stop-motion is carried by the governor for preventing a runaway in case the governor-belt should break.

A tandem compound Corliss engine of the Atlas type is sketched in Fig. 274. The steam-valve gear of this engine for both high- and

low-pressure cylinders has a through line of connecting-rods from one eccentric and a through line of connecting-rods from the other eccentric to the valve-arms on each cylinder. The governor controls

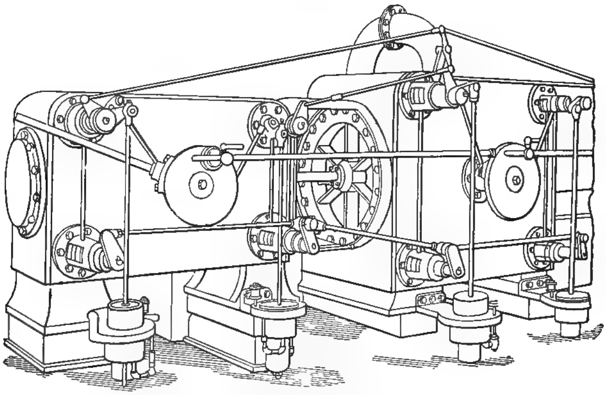


FIG. 274.—Tandem Corliss engine.

the steam- valves of the high-pressure cylinder, which open on the inside or toward the centre of the cylinder.

In Fig. 275 is shown a sketch of the cylinder and valve-gear of the Corliss engine of the C. & G. Cooper design, with two wrist-plates and

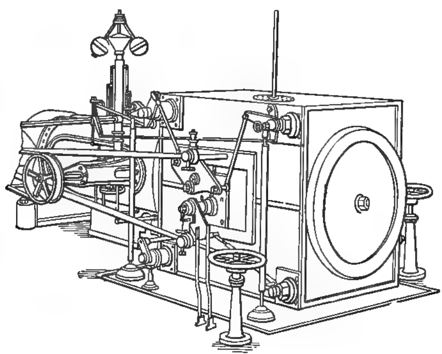


FIG. 275.—Cooper Corliss engine.

two eccentrics whose rods are transmitted to the valve-gear through two rocker-arms.

The question of a right- and left-handed engine is often raised, and

occasionally has been the subject of serious discussion. In Fig. 276 we give a sketch from good authority as to the positions of a right- and

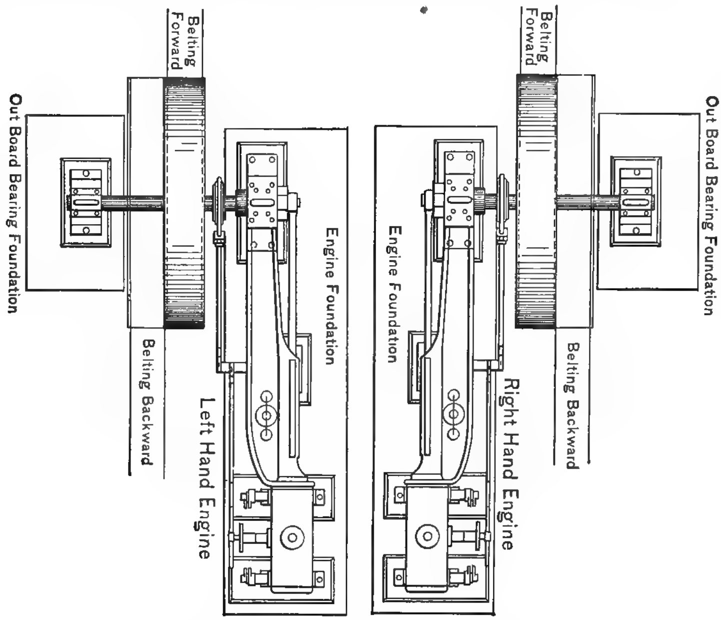


FIG. 276.—Left-handed and right-handed engines.

a left-hand engine. When standing at the cylinder and looking toward the shaft, a right-hand engine has the valve-gear on the right-hand side, and vice versa.

CHAPTER XVII

COMPOUND ENGINES

THE simple compounding of the steam-engine is a source of economy over the single-cylinder non-condensing type wherever water for condensation cannot be made available. The simple condensing-engine was formerly the most important step in the progress of steam-power, until followed by the multicylinder compound condensing type, of which simple compounding is probably the most economical method of developing power from high-pressure steam where condensing water is not available. This system is now being largely adopted for locomotive service and for localized power as a means of greater expansion from higher pressure and with high speed. The loss by cylinder-condensation is lessened by compounding, as is approximately shown in the following table:

TABLE XXXV.—PERCENTAGE OF LOSS BY CYLINDER-CONDENSATION.

Percentage of stroke completed at cut-off.	Simple engines.	Compound engines, high-pressure cylinder.	Triple-expansion engines, high-pressure cylinder.
5	42	..	..
10	34	26	..
15	29	24	22
20	26	22	20
30	22	18	16
40	18	15	13
50	14	12	10

The water-consumption in a compound engine as compared with a single-cylinder non-condensing engine is shown approximately in the following table, general conditions only being considered:

The value in compounding the expansion of steam for power has been amply shown in a practical way by the experience of the past two decades, in which its multiple effect reached its fourth stage; further advance in this line seems impracticable from the near safety-limit of boiler-pressures to the tensile strength of material for practical and economic construction.

TABLE XXXVI.—WATER-CONSUMPTION IN COMPOUND AND SINGLE-CYLINDER ENGINES.

Cut-off, per cent.	Initial pressure by gauge.		Mean effective pressure gauge.		Feed-water per indicated horse-power per hour, pounds. Compound engine.	Feed-water per indicated horse-power per hour, pounds. Single cylinder.
	High-pressure cylinder, pounds.	Low-pressure cylinder pounds.	High-pressure cylinder, pounds.	Low-pressure cylinder, pounds absolute.		
10	80	4.0	11.67	2.65	16.92	29.88
	100	7.3	15.33	3.87	15.00	25.73
	120	11.0	18.54	5.23	13.86	21.60
20	80	4.3	26.73	5.48	14.60	25.68
	100	8.1	33.13	7.56	13.67	23.77
	120	12.1	39.29	9.74	13.09	21.86
30	80	4.6	37.61	7.48	14.99	26.29
	100	8.5	46.41	10.10	14.21	24.68
	120	11.7	56.00	12.26	13.87	23.07

The liability to self-destruction in both boiler and engine is very great, as is also their cost at the highest pressure now in use—say, 200 pounds per square inch.

The experiments of Perkins in England many years since with steam at 250 pounds pressure for power purposes, ended in practical failure, and was followed by Reed and others in the United States with like results. Steam at 1,000 pounds pressure was used by Perkins, and was repeated by the author in New York with a Perkins gun, with practical failure as to its merit; and all these lessons are probably lost to the present ambition of the engineering world.

Below a limited high pressure the economy of compounding or multiple expansion is found in a reduction in the size of parts—making a lighter and less costly engine for a given power; giving in a two- or three-crank engine a more uniform twisting or turning moment, with smaller strains in the engine; enabling a smaller fly-wheel and its straining moments, and resulting in a more uniform motion where a fly-wheel is not used—and in making the range of temperature less in each cylinder, and thereby lessening the total loss in steam per indicated horse-power.

The highest efficiency of the compound non-condensing engine requires a proportion of cylinder-volumes, initial pressure, and cut-off that will give the terminal pressure in the low pressure at from 2 to 3 pounds above the atmosphere.

The compound non-condensing engine has two phases in its action, one of which is a direct discharge from the high-pressure cylinder to the low-pressure cylinder, as in the Woolf type; the other has a receiver between the cylinders. Of the first, the Westinghouse, Buckeye, and Ball engines are examples; but these may be made condensing by a small change in cylinder-volumes.

The relative areas of the high- and low-pressure cylinders for non-condensing and condensing compound engines of the Buckeye Engine Company are here given for various pressures in their tandem type.

TABLE XXXVII.—CYLINDER PROPORTIONS FOR NON-CONDENSING AND CONDENSING COMPOUND ENGINES—TWO CYLINDERS.

Steam.	Non-Condensing Compound Engines.		Condensing Compound Engines.		
	150 pounds.	125 pounds.	150 pounds.	125 pounds.	100 pounds.
Small engines.........	1 to 3.71	1 to 3.20	1 to 4.30	1 to 3.71	1 to 3.10
Large engines.........	1 to 3.64	1 to 3.06	1 to 4.00	1 to 3.64	1 to 3.06

An example of the tandem compound high-speed engine is illustrated in Fig. 277, and represents a vertical section of the Harrisburg four-valve type with Corliss valves. The steam-valves are controlled

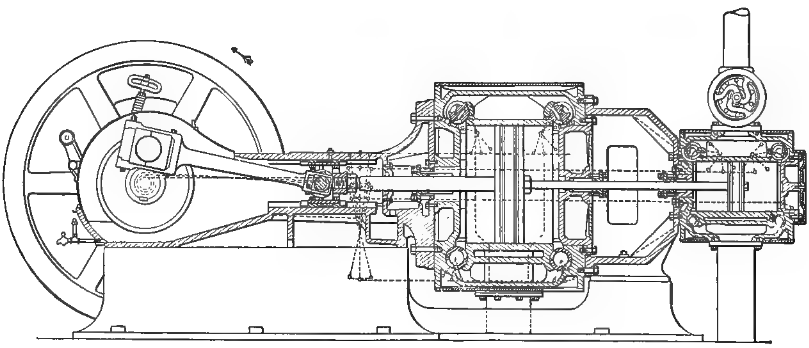


FIG. 277.—Harrisburg tandem compound engine.

by a fly-wheel governor and movable eccentric, while the exhaust-valves receive their motion from a fixed eccentric, which admits of a variable cut-off and positive exhaust-opening.

In Fig. 278 is shown a section of the cylinders and piston-valve of the Vauclain compound engine for locomotive service. A high-pressure and a low-pressure cylinder in a single casting are used on each side of the locomotive; the pistons are connected to a common cross-head, while a single piston-valve controls the events for both cylinders.

The action of the steam in this system is as follows: Steam is admitted outside of the piston-valve A, and, when the valve is moved to the right, enters the left end of the high-pressure cylinder. This action allows steam to exhaust from the right-hand end of the high-pressure cylinder through the hollow space B, in the centre of the valve A, to the left-hand side of the low-pressure piston, while steam on the right escapes through the exhaust-cavities C, C around the valve. At the proper time steam is cut off from the high-pressure cylinder and expansion takes place. This is followed by the closing of the exhaust on the other end of the high-pressure cylinder, which cuts off the steam in the left end of the low-pressure cylinder, and hence expansion occurs here also, while compression takes place in the right-hand end of the high-pressure cylinder. A starting-valve is used, connecting to both ends of the high-pressure cylinder, and opening to the low-pressure cylinder for starting.

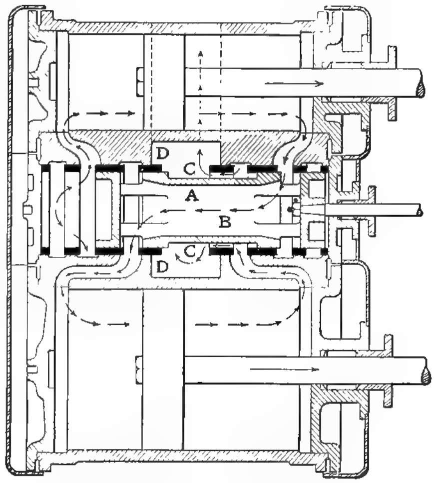


FIG. 278.—Vauclain compound cylinders.

A later design of the Vauclain type is a balanced engine in which the motions of the high- and low-pressure pistons and connections are in opposite directions, as shown in Fig. 279. The cylinders are a development of the original Vauclain four-cylinder compound type, with one piston slide-valve common to each pair. Instead of being superimposed and located outside of the locomotive-frames, the cylinders are placed horizontally in line with one another, the low-pressure

outside and the high-pressure inside the frames. The slide-valves are of the piston type, placed above and between the two cylinders which they are arranged to control. A separate set of guides and connections is required for each cylinder. The two high-pressure cylinders being placed inside the frames, the pistons are necessarily coupled to a crank-axle. The low-pressure pistons are coupled to crank-pins on the outside of the driving-wheels. The cranks on the axle are set at 90 degrees with each other, and at 180 degrees with the corresponding crank-pins in the wheels. The pistons, therefore, travel in the opposite direction; and the reciprocating parts act against and balance one another to the extent of their corresponding weight.

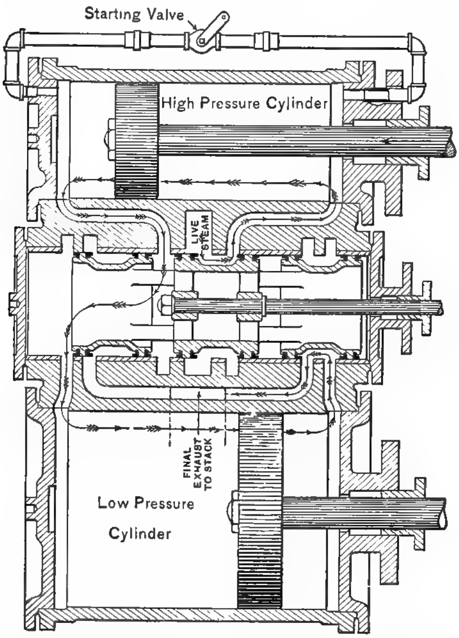


Fig. 279.—Balanced compound cylinder. Vauclain.

The distribution of steam is shown in the diagram (Fig. 279). The live-steam port in this design is centrally located between the induction ports of the high-pressure cylinder. Steam enters the high-pressure cylinder through the steam-port and the central external cavity in the valve. The exhaust from the high-pressure cylinder takes place through the opposite steam-port to the interior of the valve, which acts as a receiver. The outer edges of the valves control the admission of steam to the low-pressure cylinder. The steam passes from the front of the high-pressure cylinder, through the valve, to the front of the low-pressure cylinder, or from the back of the high-pressure to the back of the low-pressure cylinder. The exhaust from the low-pressure cylinder takes place through the external cavities under the front and back portions of the valve, which communicates with the final exhaust-port. The starting-valve connects the two

live-steam ports of the high-pressure cylinder, to allow the steam to pass over the piston.

In Fig. 280 are shown a vertical section and a cross-section of a convertible compound engine, the Flinn type for steam-automobiles and -trucks. Steam enters at the centre of the high-pressure steam-valve, and when the intercepting valve is in the position shown in the left cross-section, it can pass from the high-pressure chest directly to the low-pressure chest, allowing both cylinders to run with high-pressure steam, the high-pressure exhausting at A into the main exhaust-chest. This gives great starting or climbing power to the vehicle. Otherwise the intercepting valve is turned to the position shown in the right-hand lower section, closing the free exhaust from the high-pressure cylinder and the live-pressure connection to the low-pressure steam-chest (compelling the exhaust of the high-pressure cylinder to enter the receiver) and to the low-pressure valve.

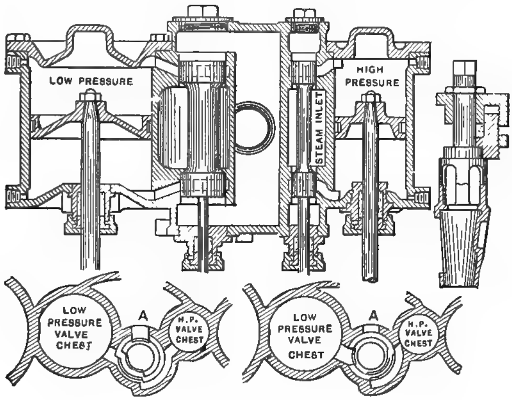


FIG. 280.—Convertible compound engine.

A non-condensing compound engine with contiguous cylinders and pistons connected to a common cross-head is shown in Figs. 281 and 282—the product of the American Engine Company. It is a high-speed engine in which simplicity and compactness have been realized to a high degree. The valves are of the duplex-piston type on a single rod, and are operated from an outside crank-pin and centrifugal governor on the outside of the fly-wheel, which method makes an automatic adjustment simultaneously for both valves.

The arrangement of pistons and cross-head will be understood by reference to Fig. 282. The cross-head here shown is designed with a view to securing the greatest strength and rigidity with the least

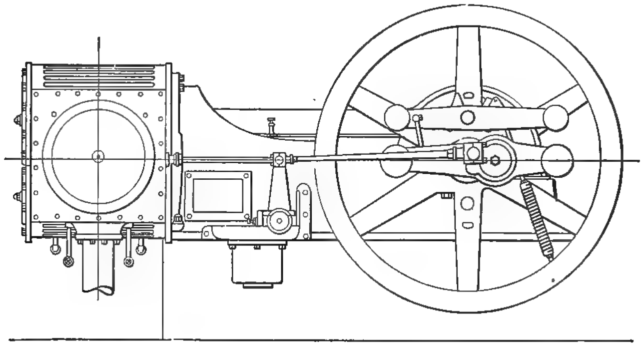


Fig. 281.—Elevation—duplex compound engine.

weight. The length is made equal to about twice the length of stroke of the engine, so that its smoothness of running is quite independent of any unequal division of work between the pistons, if such should occur; but because of the simultaneous cut-offs in both cylinders the

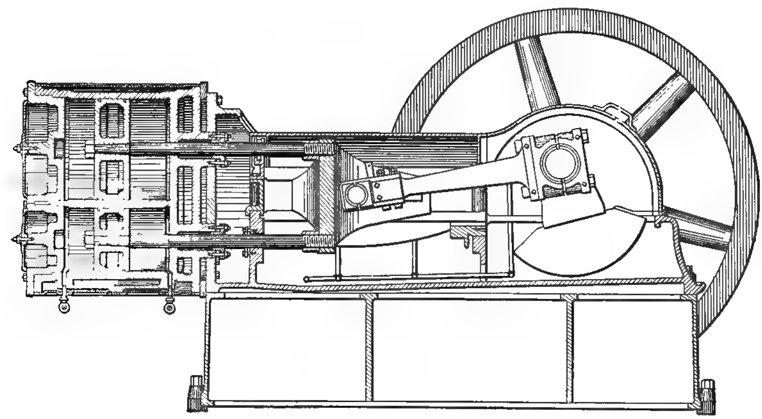


Fig. 282.—Vertical section—duplex compound engine.

work is divided almost exactly between the two pistons at all stages of load, from the simple friction load to the fullest overload capacity.

A feature of this construction which appeals to the engineer in

comparing it with the tandem compound is the fact that both pistons are as accessible as with the simple engine.

The indicator-diagrams (Fig. 283), taken from a nominally 80 horse-power engine with cylinders $9\frac{1}{2}\times15\times11$, at 275 revolutions, 100 pounds pressure, show remarkable uniformity in the division of work at greatly varying loads, which was 38 horse-power for the upper card and 85 horse-power for the lower card. Practically the same proportions are shown by the cards of the larger condensing-engine of the same type.

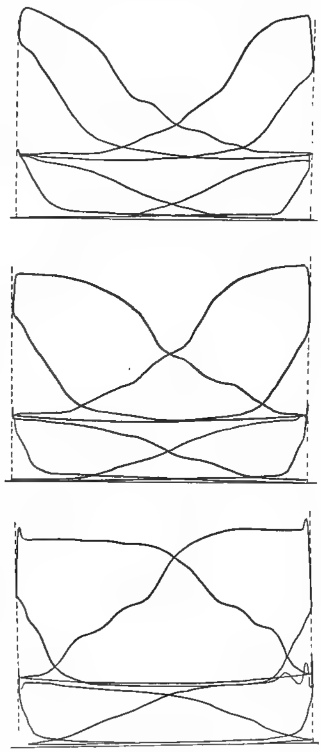


FIG. 283.—Indicator-diagrams.

A single-acting vertical cross-compound engine with cranks at 180 degrees and in which a single piston-valve set crosswise on the heads of the contiguous cylinders controls the steam-distribution, is shown in the longitudinal and cross sections, Fig. 284. It is a novel and very compact high-speed motor for all purposes, and is especially suited for direct connection to electric generators.

The steam-chest is a separate casting bolted to the cylinder-heads, and contains the steam-passages and the transfer-passage from the high- to the low-pressure cylinder. The pistons are of the trunk type, the low-pressure cylinder having an inserted trunk-bearing sleeve that encloses cushion-spaces, Y, Y of air- or steam-leakage, which is drained by the pipe and valve at *g*. The sleeve in the low-pressure trunk being of equal area with the high-pressure trunk, it serves to prevent unequal pressure in the base A and discharge at the overflow W and at the air- or steam-vent Z. The spool piston-valve H travels in a multiported liner which gives ample port-opening for high speed. The steam-chest P and the exhaust-chest T completely surround the liner. The passage X is a by-pass to the neck of the spool-valve for admitting steam to the low-pressure cylinder in starting, the liner having a set

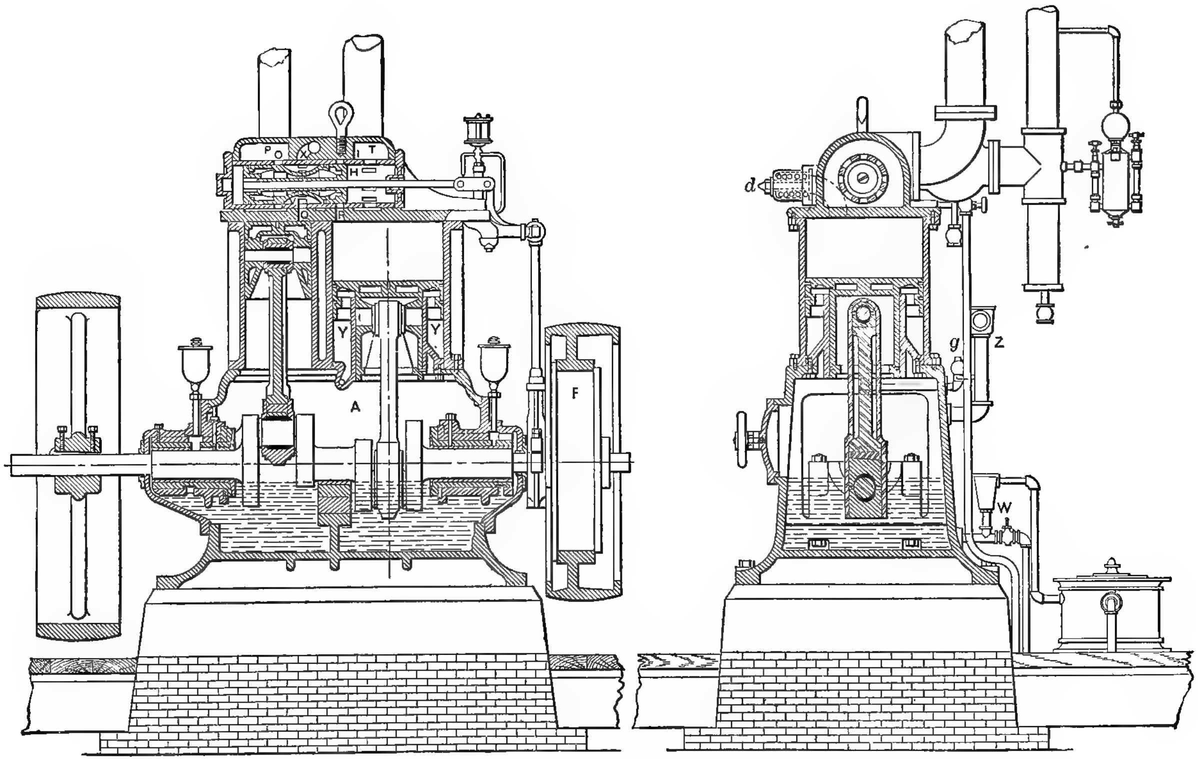


FIG. 284.—Westinghouse cross-compound single-acting engine. Longitudinal and cross sections.

of ports to meet this purpose; *d* is a relief-valve for each cylinder. The centrifugal governor is enclosed in the fly-wheel at F.

In Fig. 285 is a combined diagram of the distribution of steam-pressure in the Westinghouse vertical compound engine, taken on the same card at 90 pounds boiler-pressure. The same would be obtained if they were taken with different indicators, but with the same number of spring. The cut-off point is c in the high-pressure cylinder, and exhaust begins at e, but as there is only the small steam-chest of the low-pressure cylinder to exhaust into, the line eh is not steep. At h the low-pressure valve opens the port to that cylinder, and we get the steam rapidly exhausting from the high pressure to the low. This is

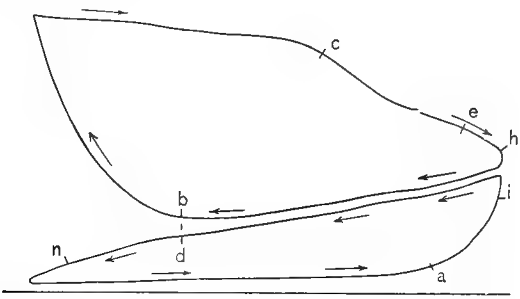


Fig. 285.—Diagram from the Westinghouse single-acting compound engine.

shown by the drop below h and the rise above i. The two pencils arrive at h and i simultaneously. The vertical line above i would meet that below h but for the resistance offered by the ports to the passage of steam from one cylinder to the other. From h to b steam is continually passing out of the high-pressure cylinder into the low, the corresponding points in the latter diagram being i and d. The cause of the fall toward d is due to the space occupied by the steam in the low-pressure cylinder, combined with that in the high-pressure cylinder *together* growing larger, and consequently the pressure must grow less. At b the exhaust-port in the high-pressure cylinder closes, and compression begins. Further, as no more steam is coming from the high-pressure cylinder, the steam in the low begins a slightly different expansion-line, dn, until at n exhaust begins. Compression begins at a and ends at i, where admission commences.

In Fig. 286 is given a diagram of a test made to determine the

steam-consumption and the mechanical efficiency and regulation of a Buffalo 12 and 18×10-inch horizontal, tandem-compound, high-speed, non-condensing engine; steam-pressure, 125 pounds. In this engine the high-pressure valve takes steam on the inside, and exhausts around the ends. The steam then passes through the cast port in the bottom of the high-pressure cylinder to a receiver-pipe on the opposite side, and then to the low-pressure valve-chest. The steam is led to an

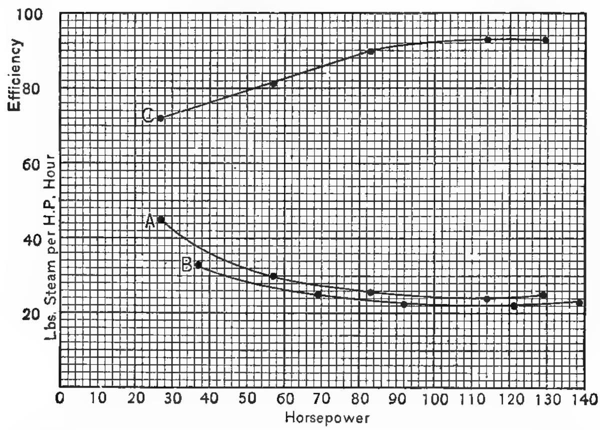


FIG. 286.—Diagram of steam-consumption and efficiency.

exhaust-outlet at the bottom of the low-pressure valve-chest. The high-pressure valve is a piston, while the low-pressure valve is a slide, with a balance-plate on the back. The two valves are moved by independent eccentrics and rods, the two eccentrics being on opposite sides of the engine.

Governing is obtained entirely at the high-pressure valve, the motion of this valve being controlled by a centrifugal or shaft governor. The engine was set to run at about 285 revolutions.

The diagram shows curves of the steam-consumption for non-condensing tests and the mechanical efficiency. Steam-consumption is figured for dry steam, the steam being at no time over 4 per cent. moisture in the main and usually less than 3 per cent. Curve A shows the relation between developed horse-power and steam per developed horse-power hour, curve B between indicated horse-power and steam per indicated horse-power hour, and curve C between developed horse-power and mechanical efficiency.

In Fig. 287 is given a diagram showing the efficiency and steam-consumption in a test of a Reeves vertical cross-compound non-condensing engine, 12 and 20 × 14 inches, and a condensing-engine $10\frac{1}{2}$ and 20 × 14; it shows the difference in efficiency and steam-consumption under like conditions of steam-pressure and valve-gear. The ratio of cylinder-volume for the condensing trial was 1 to 3.6, vacuum 24-inch, and for the non-condensing trial 1 to 2.75, and normally 160 horse-power.

In this test the power was measured by the use of a Prony brake, and the steam-consumption and efficiency, over a considerable range

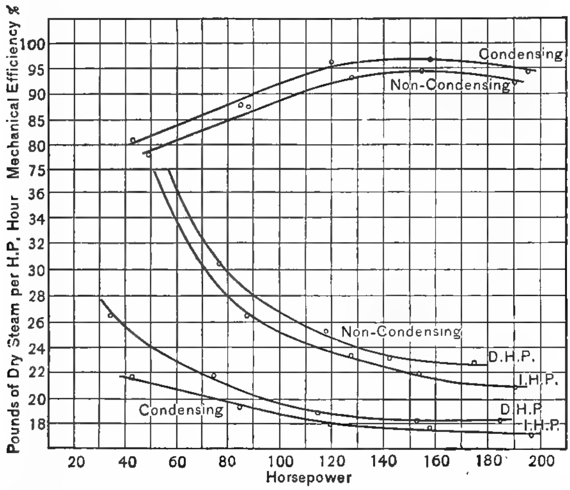


FIG. 287.—Comparative diagram of efficiency and steam-consumption in non-condensing and condensing engines.

of load, are easily seen from the curves. Two things may be noticed, however: The cut-off, even at the higher loads, does not seem to be late enough to cause any marked rise in the steam-consumption; and the mechanical-efficiency curves show a steady increase with the load at any given steam-pressure, and the efficiency falls off but slightly on overload.

RECEIVERS

A receiver is not essential to a compound tandem engine with immediate connections between the cylinders, although the usual pipe-connections operate in a small measure as a receiver. With cross-compound engines with cranks at 90 degrees, the receiver modifies the steam-expansion to a considerable extent.

The distribution of the steam in the cylinders of a tandem compound engine, at various points of the stroke, is graphically shown in the diagram (Fig. 288), neglecting clearance- and steam-passages. In the diagram ab = volume of the high-pressure cylinder; ac = volume of low-pressure cylinder = 1 to 4; the vertical line ad = initial pressure; exhaust of high pressure is at f, one-third cut-off; initial pressure of low-pressure cylinder = ag = terminal pressure of the high-pressure cylinder and takes steam to the end of the stroke; the curve gmk represents the fall of pressure in the low-pressure cylinder, and the curve fnh represents the decreasing back pressure in the high-pressure

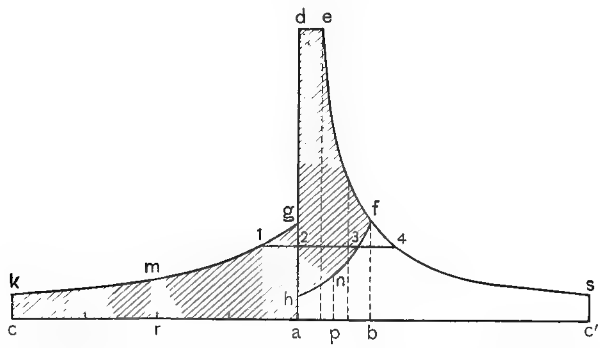


FIG. 288.—Diagram, without receiver, of tandem compound engine.

cylinder (each shaded section representing the theoretical indicator-diagram for its steam-pressure); pn = rm, and ck = ah.

In Fig. 289 is a diagram of the distribution of steam as well as the theoretical indicator-card for each cylinder, representing the shaded part. Volume of cylinders, 1 to 3; cut-off, one-half in each cylinder. Then, if the steam be admitted to the high-pressure cylinder for one-half the stroke, de = $\frac{1}{2}$ ab is the line of admission, e is the point of cut-off, and ef the curve of expansion to the end of the stroke of the

high-pressure cylinder, the terminal pressure being bf $= \frac{1}{2}$ ad. Communication is now opened with the receiver, and the pressure falls to g, the pressure bg depending on the volume of the receiver and on the pressure of the steam in it. But there is as yet no admission to the low-pressure cylinder till another half-stroke has been made. The diagram of work done by the high-pressure piston will therefore show an increasing back-pressure curve, gt, as that piston returns, till it

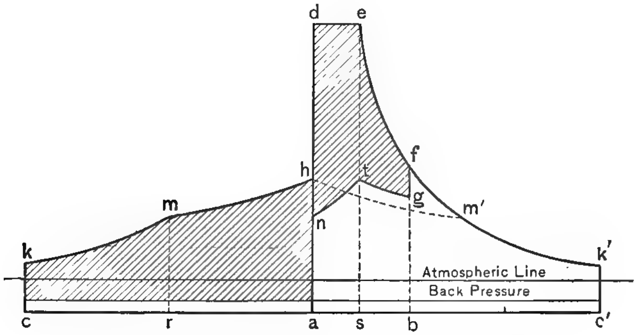


FIG. 289.—Diagram with receiver and condenser.

reaches half-stroke, when the low-pressure steam-port opens and admits steam at the initial pressure ah = st. The pressure now falls by expansion of the steam behind the low-pressure piston, the terminal pressure an in the high-pressure cylinder being equal to the pressure rm in the low-pressure cylinder at half-stroke. Cut-off now takes place in the low-pressure cylinder, and the steam expands behind the piston to ck $= \frac{1}{6}$ ad $= \frac{1}{3}$ bf, at which point it escapes to the condenser, when the pressure falls to the line of back pressure.

The volume of a receiver is relative to the volume of the high-pressure cylinder for the best distribution of the steam, and for two expansions it should be five times the volume of the high-pressure cylinder. For triple expansion the first receiver should be six times the volume of the high-pressure cylinder; for the second receiver its volume should be four times the volume of the intermediate cylinder. The variation in the receiver-pressure will be greater for a small receiver than for a large one, and also depend upon the initial pressure for any proportional size.

As an illustration of the line of pressure in the receiver of a cross-

compound engine with cranks at 90 degrees and cylinder proportions 1 to 4, the diagram (Fig. 290) represents the general conditions. In the normal running of the engine with the cut-off in the high-pressure cylinder at one-half and in the low-pressure cylinder at one-fourth stroke, the pressure will increase in the receiver from the opening of the exhaust in the high-pressure cylinder at j until the valve of the low-pressure cylinder opens at l, then falls until the cut-off of the low-pressure cylinder takes place at m, then rises until the high-pressure stroke is completed at n, and the next exhaust repeats the pressure-curve from j to l. With the cut-off at half-stroke in each cylinder, the pressure lines lm and mn will be extended, each to the half-length of the stroke.

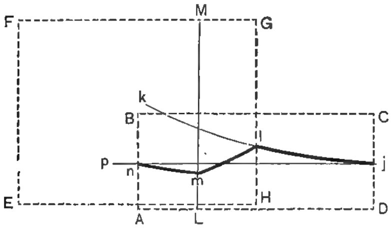


Fig. 290.—Diagram of receiver-pressure.

In Fig. 291 are shown the relative position of the pistons at exhaust in the high-pressure cylinder and cut-off at half-stroke in the low-pressure cylinder—when the receiver-pressure is low, as in the left-hand figure—and at the moment of cut-off in the high-pressure cylinder and valve-opening in the low-pressure cylinder, when the receiver-pressure is high, as in the right-hand figure.

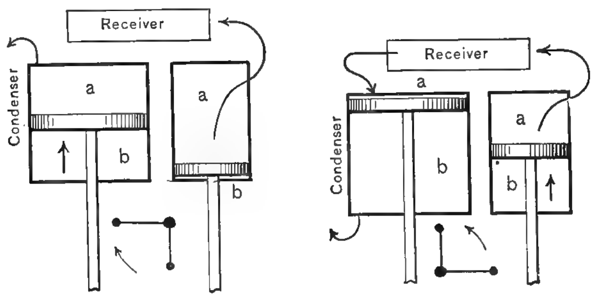


Fig. 291.—Diagram of pressures in receiver, variable.

The economy of reheating the steam in the receiver has not been satisfactorily defined by experiments made in Europe and the United States for this purpose. In a series of trials in England for reheating the steam within the receiver by a copper helical coil, negative results were obtained in nine-tenths of the trial runs. The conclusions

arrived at by these trials were that the influence of the reheater was as follows:

(a) Reducing the amount of condensation in the receiver.

(b) Raising the receiver-pressure.

(c) Raising the mean pressure throughout the engines.

(d) Increasing the speed of revolution of the engines.

(e) Increasing the dryness of the steam acting in the low-pressure cylinder.

These may be ranked as effects which are in themselves favorable to economy.

The influence of the reheater was found, however, equally marked in effects which were detrimental to economy, namely:

(a) Lowering the mechanical efficiency of the engine.

(b) Increasing the steam-consumption per horse-power developed.

Experimental tests in the United States have added no positive economical results by reheating, over a thorough felting of the receiver and connecting-pipes.

The use of superheated steam now coming into consideration and practice for all conditions of expansion, is of such importance in the economy of multicylinder engines for both land and marine service, that reheating in receivers will no doubt be left in its experimental stage in deference to the better economy of superheat.

CHAPTER XVIII

TRIPLE- AND QUADRUPLE-EXPANSION ENGINES

THE increased efficiency due to a greater range of expansion in the triple and quadruple effect from high pressures has brought steam-power to its highest degree of usefulness and economy. It is but a few years since, in the memory of old engineers, that a 10-per-cent. thermal efficiency was good practice; but improvements in metallurgy and the mechanic arts—which have made possible great advances in steam-pressure and its multiple expansion—together with the progressive experience in design, have more than doubled the power-economy of former years and brought the thermal efficiency up to 23 per cent. and the mechanical efficiency to 95 per cent., and possibly more. It is claimed that less than 1 pound of coal per indicated horse-power per hour has been obtained in test trials.

The amount of steam or its equivalent per indicated horse-power per hour varies with the pressure and cut-off at the higher pressures now in use; and its distribution in triple-expansion engines is shown approximately in the following table:

TABLE XXXVIII.—WATER-CONSUMPTION IN TRIPLE-EXPANSION ENGINES.

Cut-off, per cent.	Initial Pressure by Gauge.			Mean Effective Pressure, or Above Vacuum.			Feed-water per I. H.-P. per hour, pounds.
	High-pressure cylinder, pounds.	Intermediate-pressure cylinder, pounds.	Low-pressure cylinder, pounds.	High-pressure cylinder, pounds.	Intermediate-pressure cylinder, pounds.	Low-pressure cylinder, pounds.	
30	120	37.8	1.3	38.5	17.1	6.5	12.05
	140	43.8	2.8	46.5	18.6	7.1	11.4
	160	49.3	3.8	55.0	20.0	8.0	10.75
40	120	38.8	2.8	51.5	22.8	8.6	11.65
	140	45.8	3.9	59.5	23.7	9.1	11.4
	160	51.3	5.3	70.5	25.5	10.0	10.85
50	120	39.8	3.7	60.5	26.7	10.1	12.2
	140	46.8	4.8	70.5	28.0	10.8	11.6
	160	52.8	6.3	82.5	30.0	11.8	11.15

The record of a test of a high-duty triple-expansion pumping-engine of the Allis-Chalmers vertical type, lately erected at the St. Louis water-works, is worthy of reference for its showing of thermal and mechanical efficiency.

RESULTS OF DUTY TEST.

Duration of test	24 hours.
Diameter of cylinders	34 inches, 62 inches, and 94 inches.
Stroke of engine	72 "
Diameter of plungers	$33\frac{7}{8}$ "
Average steam-pressure at engine	140.24 pounds.
Average first receiver-pressure	26.36 "
Average second receiver-pressure	2.77 "
Average vacuum-pressure by cards	13.21 "
Average barometer-pressure	14.46 "
Average net head pumped against	238.2323 feet.
Average revolutions per minute	16.539
Piston-speed per minute	198.44 feet.
Total water pumped	20,070,690 gallons.
Total water received from engine	220,129 pounds.
Average moisture in steam	0.13 per cent.
Indicated horse-power	865.23 horse-power.
Delivered horse-power	842.69 " "
Per cent. friction	2.60 per cent.
Average moist steam per indicated horse-power per hour	10.60 pounds.
Average dry steam per indicated horse-power per hour	10.59 "
Average British thermal units per indicated horse-power per minute	201.39 British thermal units.
Mechanical efficiency	97.4 per cent.
Duty per 1,000 pounds of steam	181,068,605 foot-pounds.
Duty per 1,000,000 British thermal units	158,851,000 " "
Thermal efficiency	21.06 per cent.

As the multicompounding of steam-expansion enables the fullest advantage to be taken of the expansion from the highest pressures available—by reducing the range of temperature in any one cylinder and its initial condensation, and by utilizing its reëvaporation in a succeeding cylinder—as also the economy due to the greater total range of temperatures with the lesser extreme strains in the mechanisms—this system of steam-power has been brought to apparently the highest degree of perfection possible.

In the diagram (Fig. 292) are shown the approximate divisions in a triple-expansion engine as between temperatures and absolute

pressures from an initial pressure of 150 pounds absolute. These proportions may be varied to suit the equalized total pressure in each cylinder for any initial pressure proposed.

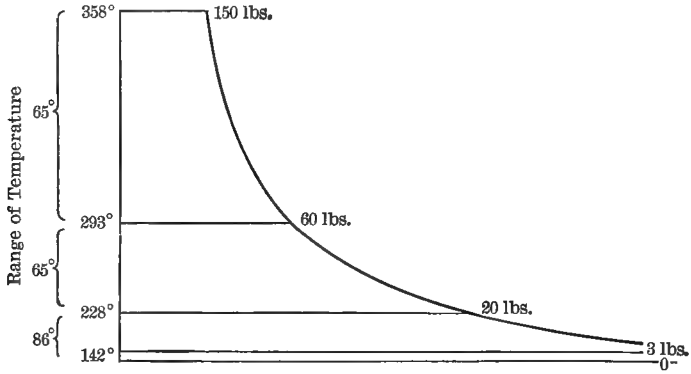


FIG. 292.—Pressures and temperatures in triple expansion.

The cylinder disposition and proportions in triple- and quadruple-expansion engines vary in a considerable degree, and are illustrated in the following figures:

A high-pressure, intermediate-pressure, and low-pressure cylinder in line with a three-crank shaft, with cranks at 120 degrees (Fig. 293).

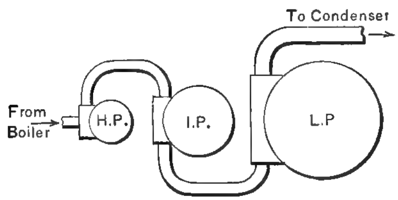


FIG. 293.—Triple expansion.

A high-pressure, intermediate-pressure, and two low-pressure cylinders on a four-crank shaft at 90 degrees, or alternating at 180 degrees (Fig. 294).

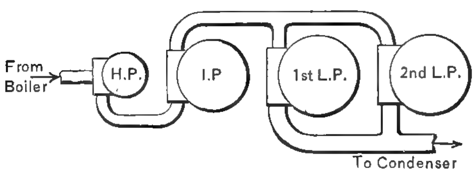


FIG. 294.—Four-cylinder triple expansion.

A quadruple-expansion engine with a high-pressure, consecutive first and second intermediate-pressure, and one low-pressure cylinder (Fig. 295). The quadruple is also built on the tandem model, with high and first intermediate tandem vertical and second intermediate and low tandem vertical in the marine service.

The relative cylinder-volumes in these engines for high initial

pressure are from 2.3 to 2.7, and are divided so as to give about 20 as the total number of expansions in a quadruple-expansion engine.

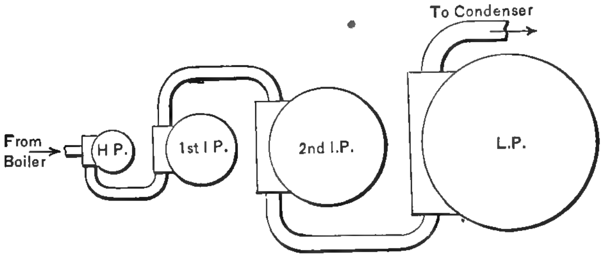


FIG. 295.—Four-cylinder quadruple expansion.

The relative cylinder-volumes in some recently designed triple-expansion marine engines of over 5,000 horse-power to each engine, arranged for one high-pressure, one intermediate-pressure, and two low-pressure cylinders, are as 1:2.7:2.6 volumes. The latest practice, as shown in the designs of the United States Bureau of Steam-Engineering for the engines of the *North Carolina* and *Montana*, has proportions of cylinder-volumes of 1:2.64:2.71, with one high-pressure, one intermediate-pressure, and two low-pressure cylinders in triple-expansion engines; and with piston-valves for all the cylinders, there being one piston-valve for the high-pressure cylinder and two each for the others.

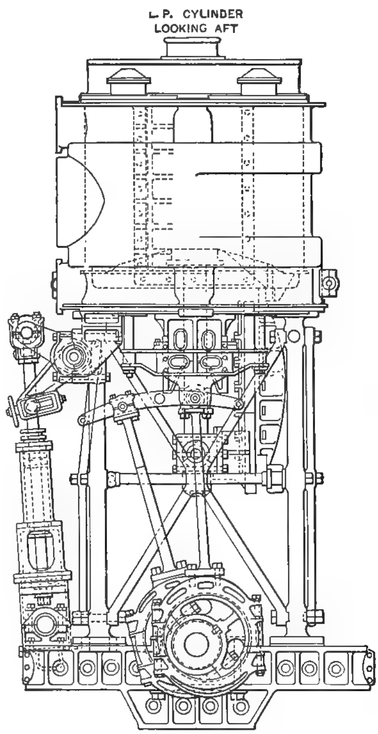


FIG. 296.—End view of triple-expansion engine. U. S Navy.

The illustrated details of these engines are shown in Figs. 296, 297, and 298, and represent results of design from the latest ideas for compactness and large power for war-ships which consist of two en-

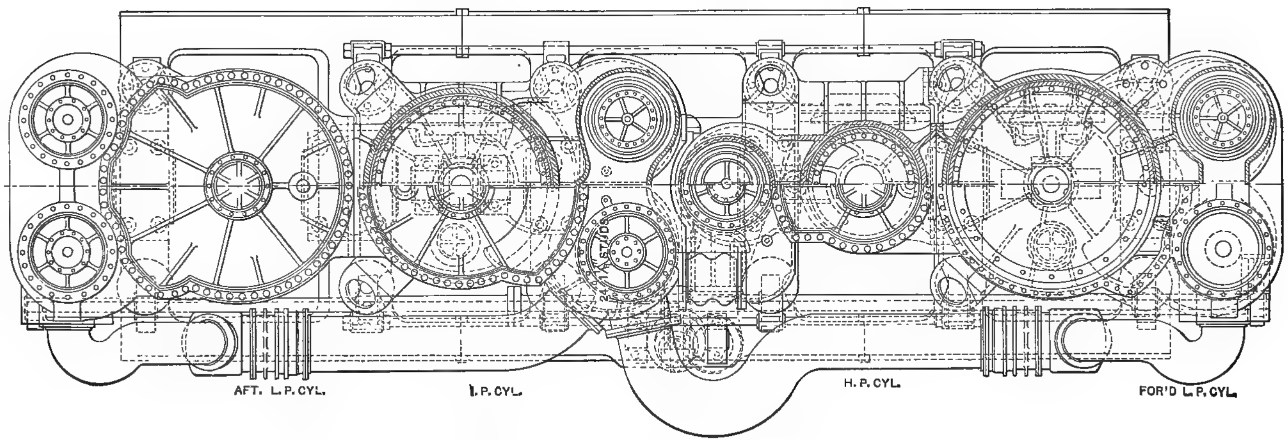


FIG. 297.—Plan of four-cylinder triple-expansion engine. U. S. cruiser *Montana*.

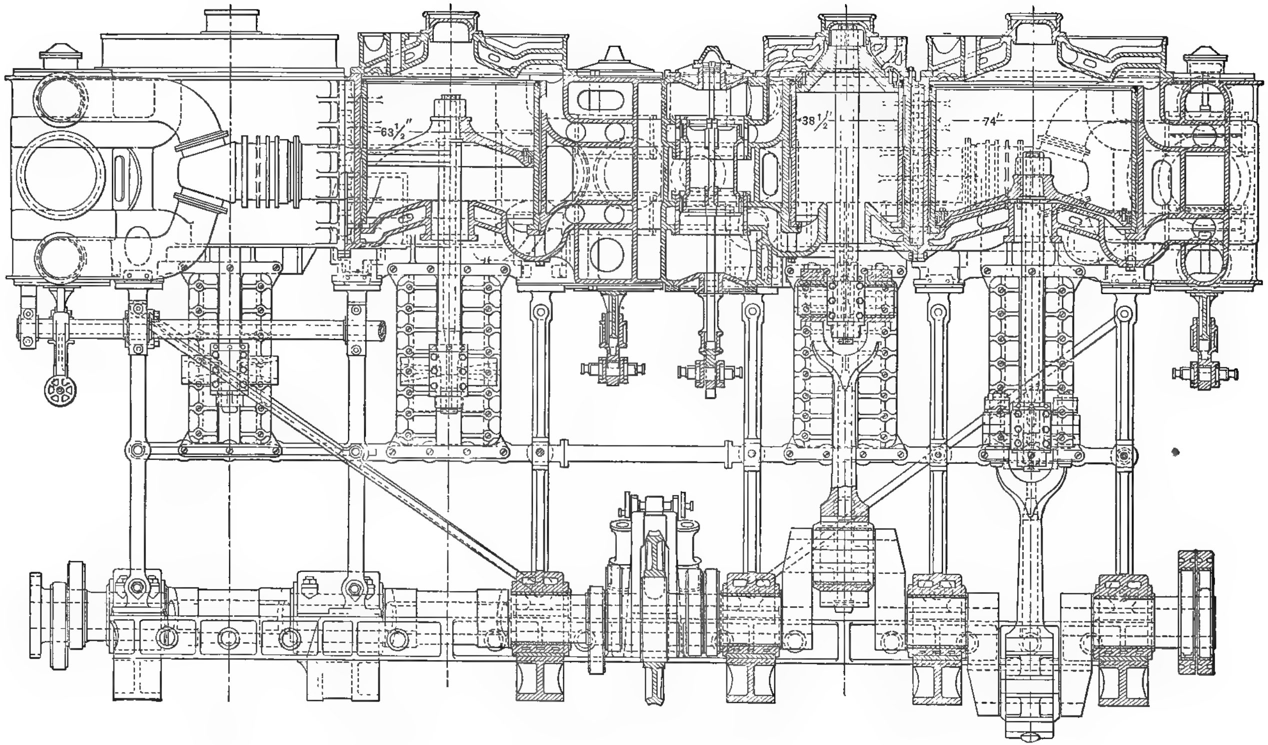


FIG. 298.—Vertical section of four-cylinder triple-expansion engine. U. S. cruiser *Montana*.

gines of 23,000 combined indicated horse-power, and with the following details of construction:

Number of cylinders................	One high-pressure, one intermediate-pressure, two low-pressure.
Diameter of cylinders...............	$38\frac{1}{2}$ inches, $63\frac{1}{2}$ inches, 74 inches each.
Stroke.............................	48 "
Piston-valves for all cylinders.	
Initial steam-pressure...............	250 pounds.
Boiler-pressure.....................	265 "
Diameter of high-pressure piston-valve	$24\frac{1}{2}$ inches.
Diameter of each low-pressure piston-valve..........................	27 "
Revolutions........................	120 per minute.

All main and valve cylinders have liners. Piston-clearance in all cylinders is $\frac{3}{8}$ inch at the top and $\frac{5}{8}$ inch at the bottom, with a probable average total clearance of about 2 per cent.

The main shaft, crank-pins, cross-head pins, and piston-rods are hollow.

Cranks of high- and low-pressure cylinders, adjacent, are 180 degrees; cranks of intermediate- and low-pressure cylinders, adjacent, 180 degrees; cranks of each pair, 90 degrees.

A novel triple compound marine engine which may be used condensing for a quadruple effect, is shown in front and cross sections

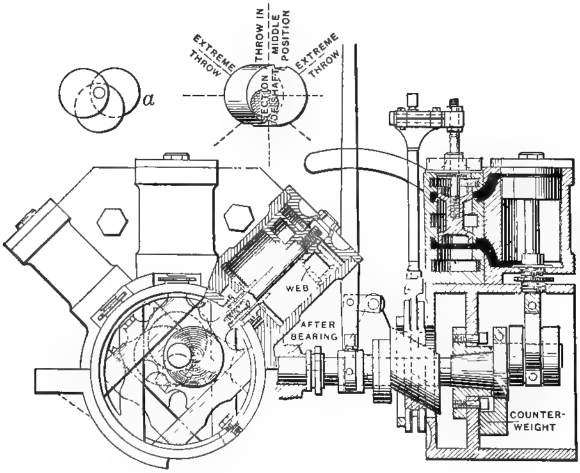


FIG. 299.—Triple compound marine engine.

in the two-part Fig. 299. The principal novelty is the three-part eccentric oscillating upon the crank-pin, and upon each of which a

strap fixed to the piston-rod of each cylinder slides in ways parallel with each piston-rod. The throw of the eccentrics and that of the crank are each equal to one-half the piston-stroke. The crank-eccentrics are set at 120 degrees, as shown at *a*. The three-piston valves are directly connected by rods to thin straps on an angularly mounted cylinder that slides on the shaft by the hand-lever for forward, stop, or reverse motion.

Piston-valves are used, taking the steam in the middle and exhausting at the ends. The steam passes from the first valve, through the triangular space between the cylinders, to the next valve-chest.

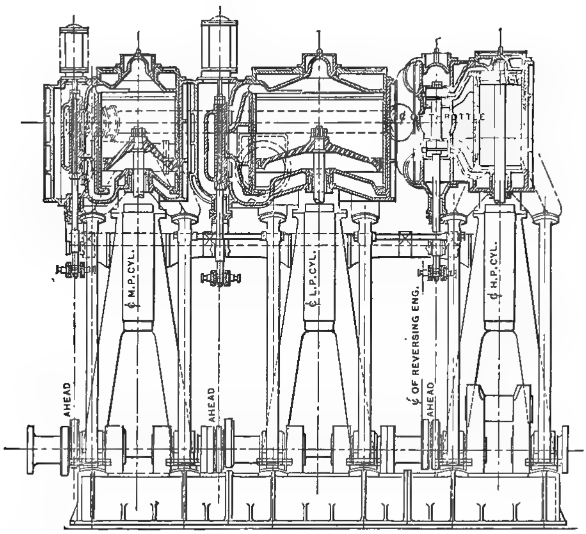


FIG. 300.—Triple-expansion marine engine. Type of the steamer *Minnesota.*

In Fig. 300 is shown a vertical section of a triple-expansion marine engine in which the high-pressure cylinder has a piston-valve, while the intermediate- and low-pressure cylinders have slide-valves. Proportion of cylinders, 1, 5, 15 in area; stroke, 48 inches; crank positions, 120 degrees; high-pressure cylinder, 23 inches diameter; intermediate cylinder, 51 inches diameter; low-pressure cylinder, 89 inches diameter.

In Fig. 301 is shown a vertical section of a triple-expansion engine with a double tandem high-pressure cylinder in which its pistons act as valves to the intermediate cylinder. The object of this is to produce

an arrangement of cylinders, steam-valves, and ports whereby the back pressure of the intermediate cylinder will not act as an opposing force on the high-pressure piston, and will also furnish full pressure of steam in the intermediate without increasing back pressure in the high. Steam enters the chamber a^3, passes through an opening between the two piston-valves, which open to the upper piston, a, when it passes the bottom centre. The cut shows it in the act of closing.

When working as a triple expansion the valve closes when the piston reaches the point b^2, which allows the steam to enter cylinder

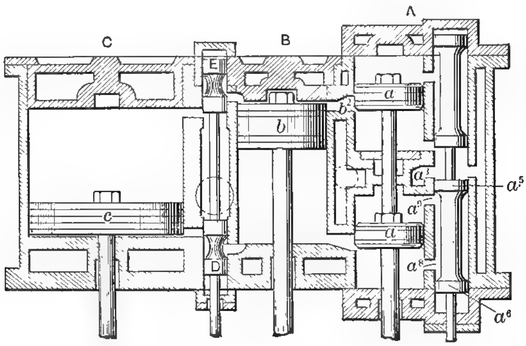


FIG. 301.—Duplex-piston triple-expansion engine.

B above piston b at full pressure, but the crank to cylinder **A** is on the quarter where it moves at its highest speed, while the piston b moves down. It will also be seen that lower piston a reaches the top of its cylinder at the same time, but instead of being in a position to exhaust as in the upper one, it will be in the position to receive through lower port a^9, valve a^5 having moved down far enough to open. The pistons a, a start, on the return-stroke, with a reversion of valve-movements. It is a curious study in steam-pressure interchange from piston port opening.

CHAPTER XIX

THE STEAM-TURBINE

THE steam-turbine, like the reciprocating engine, has a history with an inception much earlier than that of steam-expansion, and is coeval with the knowledge of steam as a power possibility. The Heron steam-motor was a reaction-turbine of which there have been several models described in the early accounts. After nearly 1,800 years since the introduction of Heron's eolipile Branca brought out, in 1629, an impulse-wheel in which a jet of steam impinged upon the flat vanes of a wheel. The principle of expansion had not yet dawned.

In the later years of the eighteenth century the principles of turbine-action came to an experimental stage, and Watt, Ericsson, Perkins, and others made trials with steam-turbines without permanent results. Up to 1901 no less than four hundred patents were issued in England on the subject of steam-turbines. In 1843 Pilbrow patented a stage-expansion steam-turbine, and Wilson, in 1848, patented the first radial-flow turbine, which in design anticipated the Dow radial-flow turbine.

In the United States the reaction-engine of William Avery had a few years of successful operation. Its rotor consisted of two hollow arms, thin and sharp like sword-blades, mounted on a hollow axis, and revolving in a dished-disk chamber. A small orifice, $\frac{1}{8} \times \frac{1}{4}$ inch, opened on the back of each blade, at the extreme end. One of its drawbacks, as claimed, was the friction-wear on the blades in cutting the exhaust-steam in the chamber.

The author's experience with an Avery turbine, erected in Buffalo, N. Y., in 1833 by his father, showed that at 1,000 revolutions per minute the stuffing-box on the hollow shaft could not be kept tight with the method of packing then in use, and that the oil in the journals and stuffing-box burned or baked by the heat of the steam and friction, and cut the bearings. Hemp and winter-strained sperm-oil were the

best materials in those days for packing and lubrication. The turbine was soon replaced by a reciprocating engine.

The Parsons type first took practical form about 1884, and the De Laval type in 1883, since which time the progress in design and the improvement in the machinery of construction have brought both types to their present efficiency and power.

In both the De Laval and Parsons steam-turbines power is generated by the impact of a jet of steam upon buckets on the periphery of a revolving disk. The essential differences between the two types of motors are these: The De Laval turbine has a single disk with several steam-jets or nozles. The nozles have divergent apertures

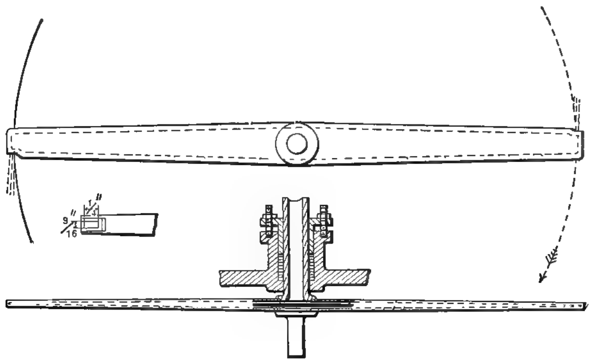


FIG. 302.—The Avery turbine.

in which the expansion of the steam takes place. The single turbine-disk revolves at a high rate of speed, say from 10,000 to 30,000 revolutions per minute, according to the size of the motor, this speed being reduced to about one-tenth on the main shaft by means of accurately cut spiral gears.

The Parsons type of turbine, on the other hand, has a series of disks mounted upon a common shaft and alternating with parallel blades fixed within the casing of the shaft. There are buckets, or cups, upon both the revolving disks and the fixed blades, the fixed buckets being reversed in relation to the moving cups. The steam, admitted first through a set of stationary blades or buckets, impinges at an angle upon the first rotating disk and imparts motion, passing thence through another set of fixed blades to the second disk upon the main shaft, and thus through the entire series of alternately

fixed and rotating buckets. The area of the passages increases progressively to correspond with the expansion of the steam as it is used on the successive disks. The expansion of steam is accomplished in the turbine instead of in the nozle, as in the De Laval motor. The buckets in a given size of Parsons turbine number about 3,000, as against about 350 in a De Laval motor of the same size.

The efficiency of the steam-turbine varies according to conditions, just as the economy of the reciprocating engine is similarly affected.

Friction is reduced to a minimum in the steam-turbine, owing to the absence of sliding parts and the small number of bearings. In one type there are practically but two bearings. The absence of internal lubrication is also an important consideration, especially when it is desired to use condensers.

As there are no reciprocating parts in a steam-turbine, and as a perfect balance of its rotating parts is absolutely essential to its successful operation, vibration is reduced to such a small element that the simplest foundations will suffice, and it is safe to locate steam-turbines on upper floors of a factory if this be desirable or necessary.

The perfect balance of the moving parts and the extreme simplicity of construction tend to minimize the wear and increase the life of a turbine and at the same time to reduce the chance of interruption in its operation through derangement or damage of any of its essential parts.

Although hardly beyond the stage of its first advent in the motive-power field, the steam-turbine has met with much favor, and there is promise of its wide use for the purposes to which it is particularly adapted. At present, however, its uses are restricted to service that is continuous and regular, its particular adaptability being for the driving of electrical generators, centrifugal pumps, ventilating fans, and similar high-speed work, especially where starting under load is not essential.

Steam-turbines are now being built in the United States in all sizes up to 5,000 horse-power. Their use abroad covers a longer period and has become more general.

The application of the steam-turbine to the propulsion of ships has produced surprising speed results. The *Turbinia*, in which the first experiments were tried in England, was a vessel 100 feet long, 9 feet beam, 3 feet draught, and 44 tons displacement. As finally equipped this vessel attained a speed of 34½ knots at Spithead in 1897,

with about 2,300 indicated horse-power. The torpedo-boat destroyer *Viper*, subsequently built for the British Admiralty, was 210 feet long, 21 feet beam, and 350 tons displacement, and a speed of 36.858 knots was developed.

The arrangement of nozles and buckets in the De Laval type of turbines has been made with nozles impinging on buckets across the periphery of the wheel, as shown in Fig. 303. The nozles are of the

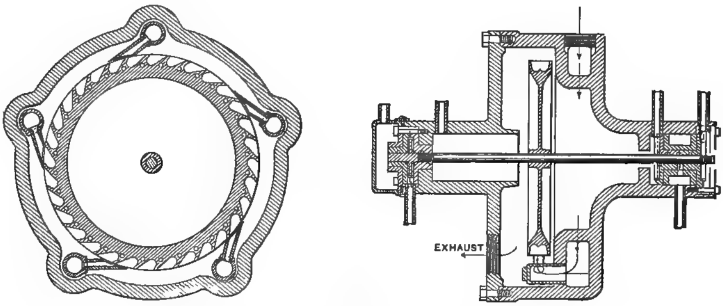


Fig. 303.—Peripheral bucket-turbine.

expanding type, as shown in Fig. 123, from which the jets of steam impinge on the edge of curved buckets of the Pelton type and discharge at the sides into the surrounding chamber. The long, slender shaft shown in the cross-section is to take up the unbalanced vibration of the disk. This model has not been credited with economical success.

The side-nozle turbine, with a number of nozles impinging upon the side at an angle of from 15 to 20 degrees against lunette buckets, and exhausting at the other side to the atmosphere or to a condenser, is the De Laval type (Fig. 304).

The wheel shown in Fig. 305 consists of a steel disk carefully balanced, and in form is very thick at the centre and made thinner as the outer edge is approached, a rim being formed at the edge, in which the buckets are mounted. A hub is formed on either side at the centre, in which is mounted the shaft, as shown. The shaft is formed of two separate pieces in the larger sizes, this form and method of mounting having proved to be the most flexible. The buckets are separate forgings of steel, held in the wheel by a bulb-shank fitting into a corresponding slot milled in the wheel.

This shank is made a driving-fit, which serves to hold the buckets

in place. The surface of these buckets against which the steam issues is not finished, but retains the hardened scale formed by forging, presenting a most excellent wearing surface. In case renewals are necessary, they can be made at small expense and in a very short time. The nozles are made of bronze, and so designed for the different steam-pressures and vacuums that they permit the free expansion of the steam. When properly proportioned for a given initial and terminal pressure, the steam as it leaves the ends of the nozles assumes a parallel form of jet, and for this reason it is not found necessary to place the nozles close to the buckets, the loss through dissipation of energy between them being so small that it can be ignored. The amount of divergence in the nozle varies considerably for different initial and terminal pressures. As the nozles do not really confine the steam, but simply prevent outside influences from affecting the free expansion, there is no wear on them, and they last for years. As the steam is expanded in these nozles to the exhaust-pressure

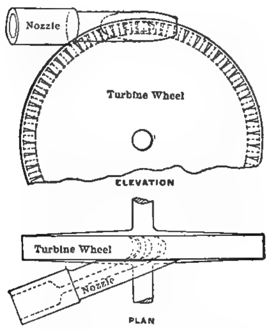


Fig. 304.—Side-nozle De Laval turbine.

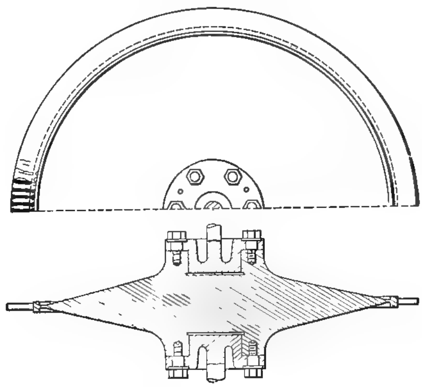


Fig. 305.—Section of De Laval turbine.

under which the turbine operates and before coming in contact with the wheel, the pressure on both sides is the same, thereby preventing any end-thrust, and the acting and reacting forces of the steam as it strikes and leaves the vanes of the wheel are substantially the same, owing to the shape of the buckets and the angle at which the steam strikes them.

The turbine-wheel revolves on its own centre of gravity by means of the flexible shaft mounted on bearings, and the floating bearings

are really metallic packing, preventing the leakage of exhaust-steam when running non-condensing, and the entrance of air into the exhaust-chamber when operated condensing. The steam, after passing through the wheel, goes direct to the exhaust-pipe in the exhaust-chamber, the space on either side of the wheel being in free communication with this exhaust-chamber. As the rotative speed of the turbine-wheel is high it is necessary to have some means of reduction in order to apply it at low speeds, and this is accomplished by means of double spiral gears of small pitch.

The system of governing consists of a balanced double-beat poppet-valve, controlled by a governor of extremely simple design and of the centrifugal type.

The steam in this type of turbine performs its work by utilizing the velocity-energy of the steam by expanding it before reaching the moving or working part of the machine. This is done through the agency of nozles of definite design, so placed as to direct the steam, after it has been expanded, against the vanes or buckets of a wheel mounted on a flexible shaft with which it rotates. With these features understood the simplicity and power of the turbine can be fully appreciated, the unusual capacity for so small a machine being due to the great speed at which it rotates. (See Chapter X on nozles and steam-velocity.)

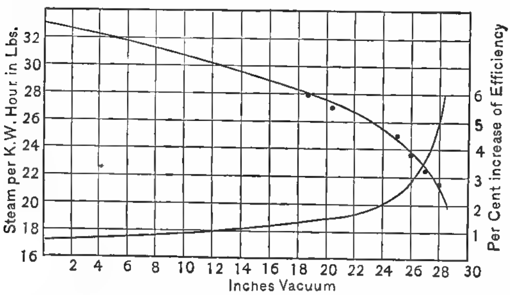


FIG. 306.—Turbine-efficiency with increase of vacuum.

The tremendous velocity which steam assumes in expanding from ordinary boiler-pressures to a vacuum—3,000 to 4,000 feet per second or 35 to 45 miles per minute—makes the use of a single wheel impracticable for turbines of large power. The stresses set up in the material of the wheel by centrifugal force prevent the employment of the peripheral speeds necessary for a satisfactory efficiency, and, except in a few cases, gearing is necessary to reduce the speed to a point where direct connection can be adopted.

The diagram (Fig. 306) shows the decrease in pounds of steam used

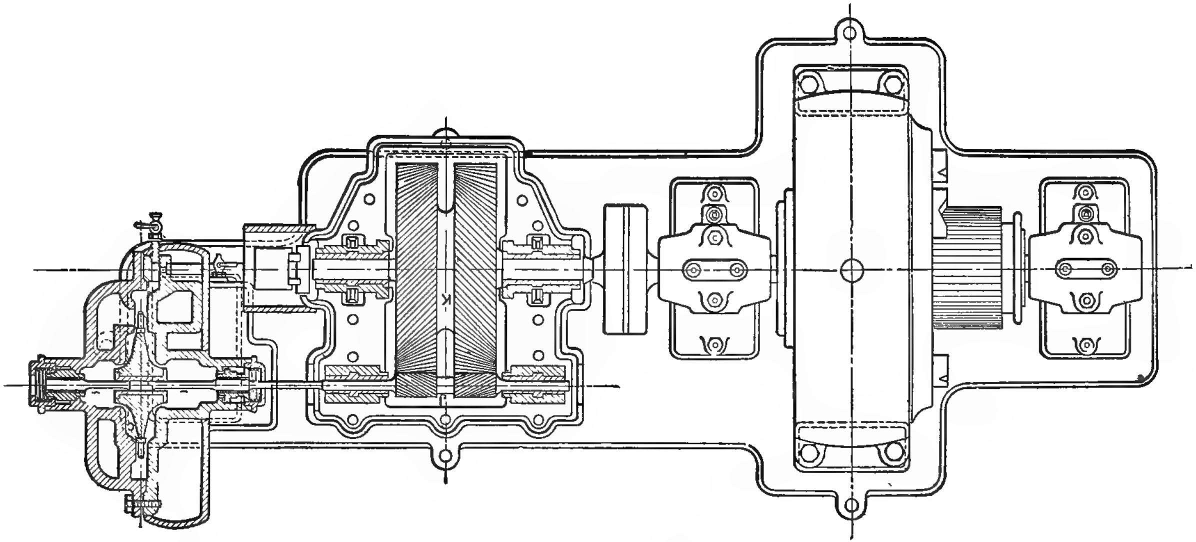


FIG. 307.—De Laval steam-turbine.

Connected to a multipolar dynamo through double spiral gears, 1 to 10; 266 B. H. P., 155 pounds steam-pressure (using 17 pounds steam per B. H. P. hour); vacuum, 25.5 inches; 9,000 revolutions per minute.

per kilowatt hour and the percentage increase in efficiency by an increase of vacuum in a steam-turbine. It shows the value of a high vacuum.

One reason why the vacuum is of particular value to a turbine is the reduction which it effects in the windage. A top will spin for a remarkably long time in a vacuum. If it had to spin in an atmosphere of compressed air or high-pressure steam, its motion would last for a comparatively shorter time. At the high speed at which the turbine is run the frictional resistance in the exhaust-pressure steam must be considerable, but in the less dense atmosphere of the vacuum much less of the energy absorbed is used in overcoming friction, and adding to the resulting power of the motor.

In Fig. 307 is illustrated a recent plan of a De Laval steam-turbine with a double compensating spiral gear and connection to a multipolar dynamo.

We illustrate some of the many forms or models in which experimental trials have been made that have, as far as the author knows, not brought out economical results in their practical development. Fig. 308 shows two views, and a section of the blades, of a Dow steam-turbine, in which two disks fixed to a shaft have on their face a series of circular grooves and tongues, meshed with a pair of fixed disks with grooves and tongues, as shown in the small section. The tongues on the revolving disks are cut across at short distances in a slanting direction. The tongues on the stationary disk are cut in the opposite direction. The steam passes to the centre hub, and is forced through the openings across the tongues, giving motion to the disks and shaft. The radial passage of the steam through blades of varying velocity seems a bar to efficiency.

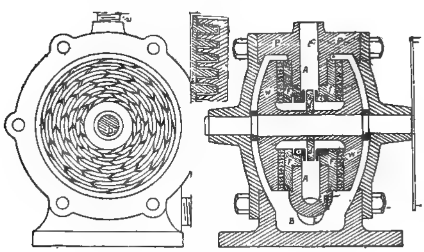


Fig. 308.—Dow steam-turbine.

Another curious modification in construction, the Wilkinson steam-turbine (Fig. 309), consists of two rim-pocketed disks running against the disk-surfaces of a shell with oblique steam-ports. The disks are feathered on the shaft, and held against the faces of the shell

and the steam-pressure by springs. A groove around the shell opposite the pockets allows the steam to pass around to the exhaust-pipes. The shape of the steam-pockets and -ports, *m*, *n*, in the rims of the disks is shown in the section at the right.

An experimental turbine by Parsons of the radial impulse type is shown in Fig. 310, in which a series of disks are fixed on a shaft with

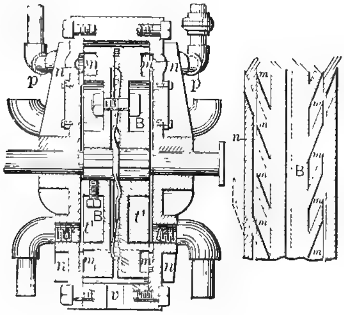


FIG. 309.—Wilkinson steam-turbine.

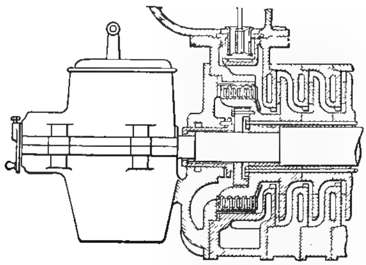


FIG. 310.—Parsons steam-turbine, early type.

intersecting disks on the shell. The face of the shaft-disks has several small blades set at an angle with the radius. The outside fixed disks have a similar set of blades interlocking with the revolving blades and set at a contrary angle. The steam passes from the valve to the inner edge of the first fixed disk, then outward through the blades, and returns through the vacant space of the next pair and outward again.

This form of the Parsons turbine is an improvement on the principle of the Dow type by multiple effect, but is still inefficient as compared with the later types of axial steam-flow.

THE MULTISTAGE STEAM-TURBINE—PARSONS TYPE

In Fig. 311 is shown a sectional view of one of the earlier models of the Parsons turbine. The steam is admitted to the chamber A, encircling the cylinder, from the governor-valve, and passes along to the right through the turbine-blades, which deflect it in one direction, thence striking the moving blades of the turbine, which deflect it in the opposite direction, and so on. In this way the current of

steam impinging upon the moving blades drives them around. The areas of the passages increase, progressing in volume corresponding with the expansion of the steam. On the left of the steam-inlet are revolving balance-pistons, C, C_1, C_2, one corresponding to each of the cylinders in the turbine. The entering steam at A presses equally against the revolving part of the turbine and against the first balancing-piston. When it arrives at the passage E it presses against the next larger section of the revolving part of the turbine and also against the next larger balancing-piston, connection between the two being secured by the passage F. Similarly, the passage G permits the balancing of the largest section of the turbine. By a proper arrangement of these balancing-pistons there is no end-thrust upon the

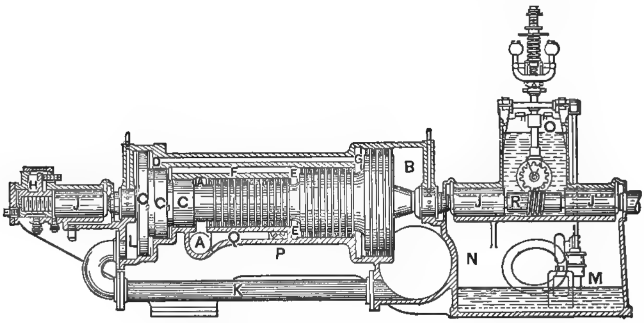


FIG. 311.—Parsons steam-turbine.

turbine-shaft at any load or steam-pressure. The thrust-bearing at H, on the extreme left, is to take care of accidental thrusts that may arise and also to retain the alignment of the shaft accurately so as to secure the correct adjustment of the balance-pistons.

Since these balance-pistons never come in mechanical contact with the cylinder in which they turn, there is no friction. The thrust-bearing is made of ample size and is subject to forced lubrication.

The pipe K connects the chamber back of the balance-pistons with the exhaust-outlet, so as to insure the pressure being equal at the two ends of the turbine.

The bearings J, J are peculiar in construction. Each consists of a gun-metal sleeve prevented from turning by a loose-fitting dowel-pin. Outside of this are three cylindrical tubes having a small clearance between them. These small clearances fill up with oil and permit a

slight vibration of the inner shell, while at the same time restraining it from too great a movement. The shaft therefore actually revolves about its axis of gravity instead of its geometrical axis, as would be the case with the bearings of the usual rigid construction. In case the shaft is a little out of balance the journal thus permits it to run slightly eccentric. The forms of the rotating and stationary blades are much like those of the Curtiss type, which are detailed in Fig. 312.

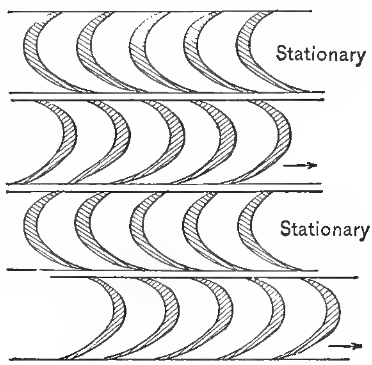


FIG. 312.—Stationary and running blades.

The casing of the turbine is lined with disks of blades curved in reverse of the blades on the rotor; all of their surfaces are of approximately parabolic form, as shown in the cut.

In Fig. 313 is represented a vertical section of the later Parsons turbine as built by the Westinghouse Machine Company. The steam from the governor-valve V enters the neck of the rotor at A through ports around the shell, and passes to the left through the successive disks of stationary and revolving blades. The area of the passage between the blades is continually enlarged to meet the increasing volume of steam by its expansion, by increasing their length, and in stepping up in area by enlarging the diameter of the rotor, until finally it is exhausted into the chamber B and into the condenser.

By this traverse of the steam there are the initial pressure upon one end of the series of rotating blades and a vacuum on the other, the difference tending to press the rotor toward the low-pressure end. This thrust is counterbalanced by a series of balancing-disks, P, P, P, equal in diameter to the respective sections of the drum. The steam enters between the smallest of these disks and the first ring of blades, and tends to push the disk to the right as much as the blade to the left, and from the chamber, before each enlargement of the drum, an equalizing-pipe or -passage, E, leads to the corresponding balancing-disk. A similar pipe connects the vacuum-chamber with the back of the largest disk, so that the pressures are effectually balanced. The balancing-disks are finely grooved on the rims and run in the

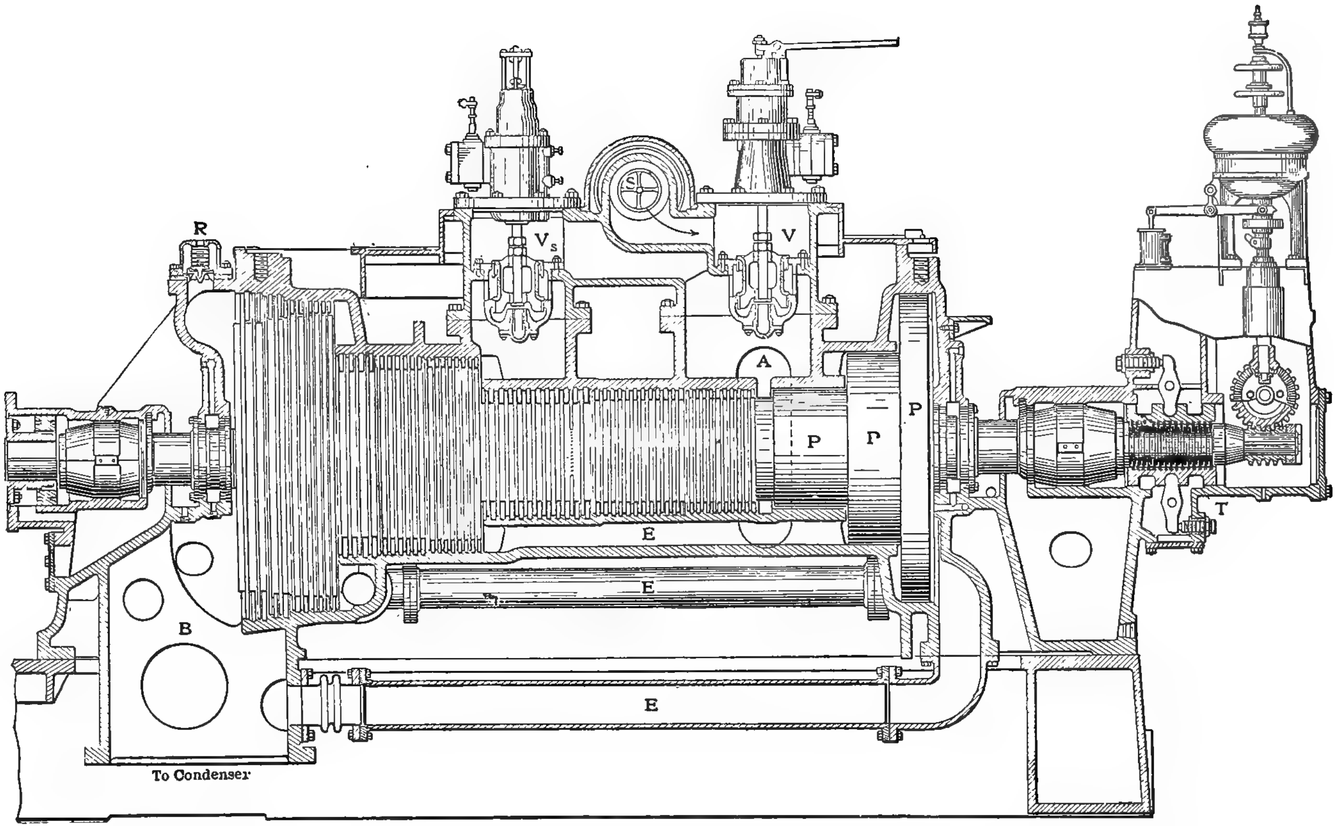


FIG. 313.—Vertical section of the Parsons steam-turbine as built by the Westinghouse Machine Company.

grooved pockets of the casing. A small thrust-bearing takes care of any incidental tendency to move endwise, and adjustment is provided for the relative positions of the blades.

The principle of action in this turbine is that the steam of the initial pressure is admitted upon one side of the smallest ring and, flowing through the spaces formed by the blades, impinges upon the first ring of rotating blades, giving them motion by its impact. But the pressure upon the exhaust side of the rotating blade is less than that upon its intake side, and the steam goes on expanding in the spaces between the blades and issues from them with a considerable velocity, adding its reaction effect to that of impact. Reversed in the next set of stationary blades, in which its expansion continues, it impacts upon the next ring of moving blades, and so on through the turbine, the space between the blades increasing progressively by their increasing length.

The admission-port for steam is shown at S. A secondary governor-valve (shown at V_s) from the admission-port provides for admitting high-pressure steam directly to the second expansion stage when the turbine is to carry heavy overloads or if the vacuum fails from any cause.

Regulation is accomplished by means of a constantly moving pilot-valve controlled by a flyball-governor. The governor-levers are mounted on knife-edges instead of pins, to secure sensitiveness. Speed may be regulated while the governor is in motion. This is particularly useful for synchronizing the speed of alternating-current machines operated in parallel and for adjusting their differences of load when so operated.

The pilot-valve controls the admission-valves, which are of the balanced vertical lift poppet type.

Steam is admitted to the turbine in puffs by means of a cam on the governor and a spring-operated piston, a steam-relay making about three impulses per second.

High-pressure steam is admitted at all loads, and the *admission-steam* is *not throttled* in proportion to the load. At full load the steam-puffs merge into an almost continuous flow.

The governor- and pilot-valve are operated by a worm-gearing on the main shaft. The pilot-valve has no "inertia of rest," and does not stick.

On the larger machines a speed-limit governor is arranged to instantly shut off the steam-supply whenever a predetermined limit of speed above normal is reached.

Frictionless glands at the ends of the stator prevent the admission of air or the escape of steam.

The rotating disks revolve within the stator with a close fit but not in contact. The adjacent surfaces are provided with frictionless packing-rings. These offer a devious path for the steam, and leakage past them is inappreciable.

A flexible sleeve-coupling connects the turbine to its generator.

Oil for the turbine- and generator-bearings is raised by a small plunger-pump from a main reservoir in the bedplate and circulates by gravity. It is cooled by water-coils.

The Westinghouse-Parsons turbine utilizes the full steam-energy and does this at rotative speeds well within commercial requirements. These speeds do not exceed 3,600 turns a minute for the 400-kilowatt unit. For the larger units the number of turns is less. The steam is also robbed of all power of erosion by having its velocity gradually reduced as it passes through the turbine.

TABLE XXXIX.—GIVES THE EFFICIENCY-TESTS OF A 500-KILOWATT, WESTINGHOUSE-PARSONS TURBINE, AND SHOWS THE RELATIVE STEAM-CONSUMPTION FOR SATURATED AND SUPERHEATED STEAM AND FOR VARYING VACUUM.

TEST.	LOAD.		STEAM.			STEAM-CONSUMPTION.	
No.	Proportion of capacity.	B. H.-P.	Pressure, pounds.	Quality.	Vacuum, inches absolute.	Total pounds per hour.	Pounds per B. H.-P. hour.
			Saturated steam.				
1	½	396.0	151.2	99.47	28.03	5,908	14.92
2	¾	584.3	152.6	99.50	28.03	8,211	14.05
3	Full	762.3	153.2	99.45	27.70	10,429	13.68
			Superheated steam				
5	Full	763.9	153.3 (105.2° F. superheat.)	....	28.00	9,334	12.22
			Reduced vacuum.				
6	Full	722.9	148.8	99.53	26.03	10,781	14.91
4	1½	1,145.5	142.6	99.58	26.30	10,429	15.08
7	Full	678.7	148.9	99.73	24.10	10,764	15.86

An efficiency-test of a 1,250-kilowatt turbine of the above type consumed 27 pounds of steam, without vacuum or superheat, per brake horse-power and 890 brake horse-power load, and 24 pounds at 1,260 brake horse-power load, 150 pounds initial pressure.

The governor-valve used on the Parsons turbine varies in construction somewhat with the different builders of steam-turbines, and may be called properly a relay or vibrating valve. It consists essentially of a double-beat or balanced valve operated by a small piston and spring, with its opening and vibration both operated from the governor. In Fig. 314 is illustrated the vibrating valve-gear, in which the double-beat valve V is shown closed. The spindle of this valve projects upward and carries a small piston, B, which is enclosed in a cylinder and held in its lowest position by means of a spiral spring, F. In the bottom of the cylinder there is a small hole, O, through which steam flows under the piston B when the main valve E is opened, so that the piston is lifted up and at the same time the double-beat valve V is opened. Steam can now flow into the turbine at A. The double-beat valve V will now remain open as long as there is steam below the piston B. In order to allow this steam to escape from time to time, there is another port-hole, D, which is considerably larger than the steam-inlet O. The port-hole D is kept closed by means of a small piston, G, which is periodically lifted in a regular jigging motion by the eccentric X, which is directly connected to the governor-spindle, so that steam escapes from the cylinder at D through the pipe H. The spring S now overcomes the piston B, which descends, thereby closing the double-beat valve V. Shortly after, the piston G is

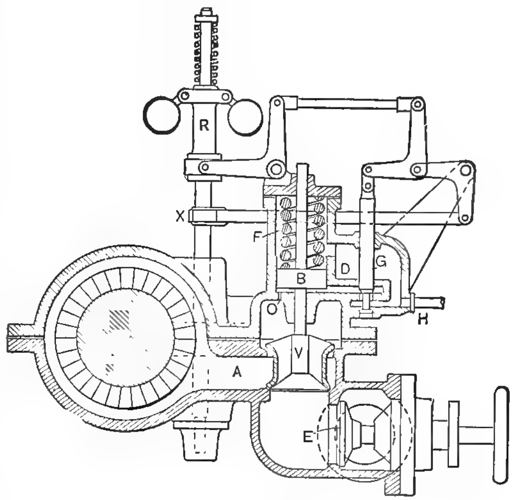


FIG. 314.—Vibrating valve-gear of the Parsons steam-turbine.

again pushed downward and the hole D closed, whereupon steam again forces up the piston B, so that the double-beat valve is once more opened. As the motion of the piston G is obtained from the eccentric X, the number of lifts bears a fixed ratio to the speed of the turbine, and the number of gusts of steam is therefore also proportional thereto. The general disposition of the governor-gear is clearly shown in the cut.

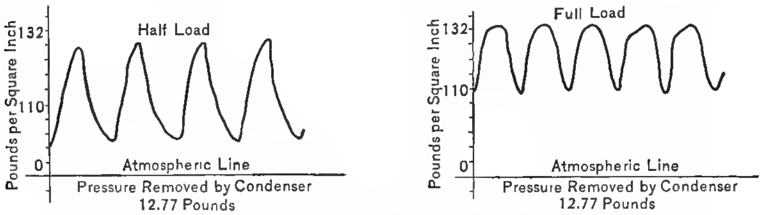


FIG. 315.—Variation in pressure of puffs at half and full loads.

Fig. 315 shows indicator-diagrams illustrating the variation of pressure due to the vibration of the relay or double beat valve at the steam-entrance A in Fig. 314. The abscissa or frequency are a function of time, and their length depends upon the speed for which the indicator-gear was set.

THE CURTISS STEAM-TURBINE

The Curtiss type of steam-turbine is assumed to be a combination of the principles of action of the De Laval and Parsons types, in that the first impact of the steam is from a series of several expanding nozles, in groups of two or three, at equal distances around the revolving wheel, directly upon the revolving blades, and from a reaction by a fixed-blade disk, and in that a further impact occurs upon the second revolving wheel-blades, the steam thus expanding through two or three stages, and terminating in the condenser. The vertical arrangement of the shaft, with the horizontal plane of motion, is one of the distinctive features of the Curtiss turbine.

In Fig. 316 is shown an elevation of a two-stage turbine with three sets of nozles equally divided around the periphery of the wheels. Each of the five or more nozles in each set is of the expanding form, with rectangular apertures extending across the wheel-blade width. Steam enters through the series of nozles, forming a broad belt of

steam, and the quantity admitted is regulated by a series of poppet-valves, one for each nozle. Regulation is effected by opening or closing these valves automatically, and for fine regulation, involving a less quantity of steam than flows through any one nozle, throttling in one nozle is resorted to. In the 5,000-kilowatt turbine there are three sets of these nozles, spaced at 60-degree angles, and the steam passes through three sets of blades into an intermediate receiver, in which the pressure is approximately that of the atmosphere. Thence it passes through a second set of guide-passages, or nozles, which expand the steam nearly to the vacuum-pressure, and the velocity of the steam is abstracted by three more sets of blades. The second row of nozles occupies the whole circumference of the wheel, to allow for the great volume of the low-pressure steam.

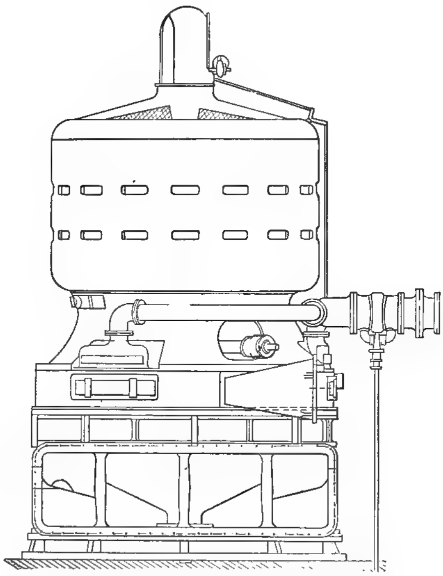


Fig. 316.—Two-stage Curtiss turbine.

In Fig. 317 are shown an outside half-view and half-section of the Curtiss turbine, with a four-stage expansion and two sets of multi-nozles. The path of the steam through the turbine is indicated in the drawing, which shows also the four stages into which the turbine is divided. From the steam-chest the steam passes through poppet-valves to the nozles which direct it upon the vanes of the first of the moving wheels of that stage, then to the stationary guide-vanes, then to the second moving wheel, from which it passes over the second set of guide-vanes to the third moving wheel. At this point the steam enters the third set of guide-vanes and undergoes a similar operation, finally passing out at the bottom of the turbine into a condenser.

In the first stage the nozles occupy but one-sixth of the circumference, and are divided into two equal sets in the turbine illustrated. In the 500-kilowatt size the nozles are all grouped together; in inter-

mediate sizes they are divided into two, and in the 5,000-kilowatt size into three, groups equally spaced. The first intermediate guides in all machines are grouped in the same manner as the nozles and occupy the same circumferential length. In the second and third stages the nozles and intermediate guides generally occupy the entire circumference.

The nozles are cored or cut passages in a cast plate, forming, in the first stage, the bottom of a steam-chest over the periphery of the turbine-wheel. The nozles of the second stage are secured to a diaphragm separating the two stages. The nozle-openings of this stage are adjusted to maintain the pressure-relation between the two stages, or to shut off the second stage entirely when the turbine is run non-condensing. This adjustment, which remains permanent for an approximately fixed condition of load, is secured by means of a register-ring rotating through a small angle. The nozles of each stage are so proportioned that the steam, when it strikes the blades, has a pressure but slightly above the exhaust, and this pressure is reduced in passage through the wheels and guides to about the exhaust-pressure.

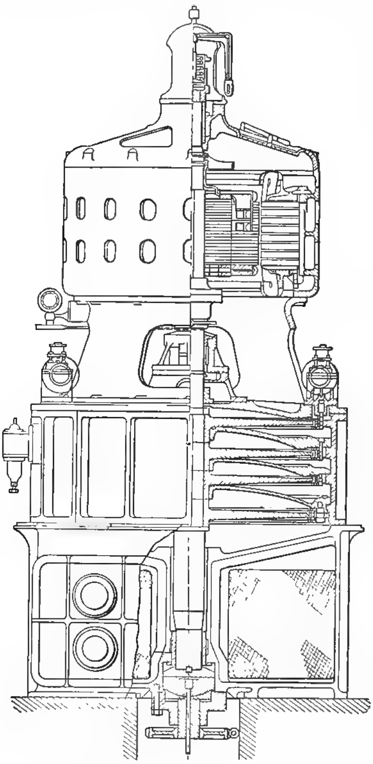


FIG. 317.—Half-section of Curtiss turbine.

In Fig. 318 is shown the arrangement of the steam-chest, valves, nozles, and moving and stationary blades for a three-stage turbine. The most vital points in a steam-turbine are the buckets, since they, and the spaces between them, must be shaped exactly right to give the correct direction of flow and highest mechanical efficiency, and also to provide for the progressive expansion of the steam. The buckets of the Curtiss turbine are cut out of the solid metal by special bucket-cutting machines. For the smaller sizes

of wheels the blades are cut from the disks comprising the wheels, and for the larger sizes the buckets are cut from segments of steel and then bolted around the periphery of the disks.

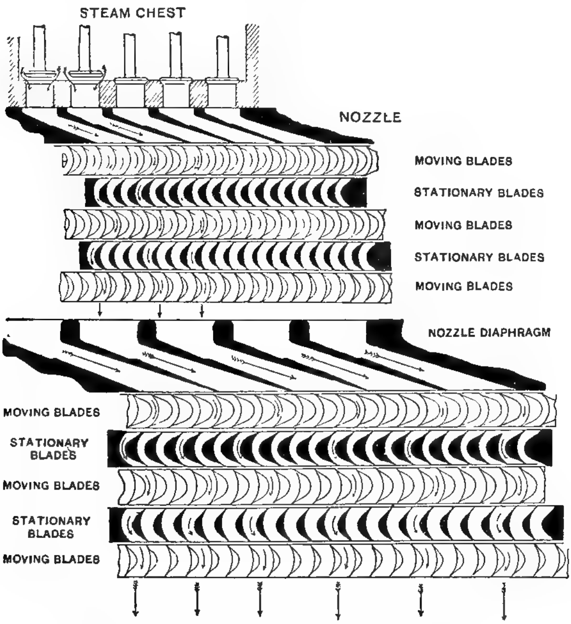


FIG. 318.—Arrangement of nozles and blades.

Fig. 319 shows a bucket- or blade-segment, with a rim of steel riveted on and enclosing the outer openings of the curved passages in the buckets.

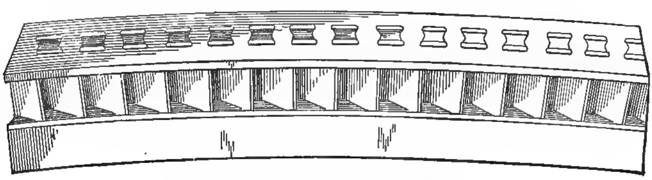


FIG. 319.—Bucket-segment.

In Figs. 320 and 321 are shown the elevation and plan of the complete installation of the 2,000-kilowatt Curtiss turbine at the St. Louis Exposition. The plant consists of a two-stage turbine, with

two sets of ten nozle-ports on opposite sides and an electric control of the nozle-valves; a small turbine-driven exciter-dynamo, a vacuum

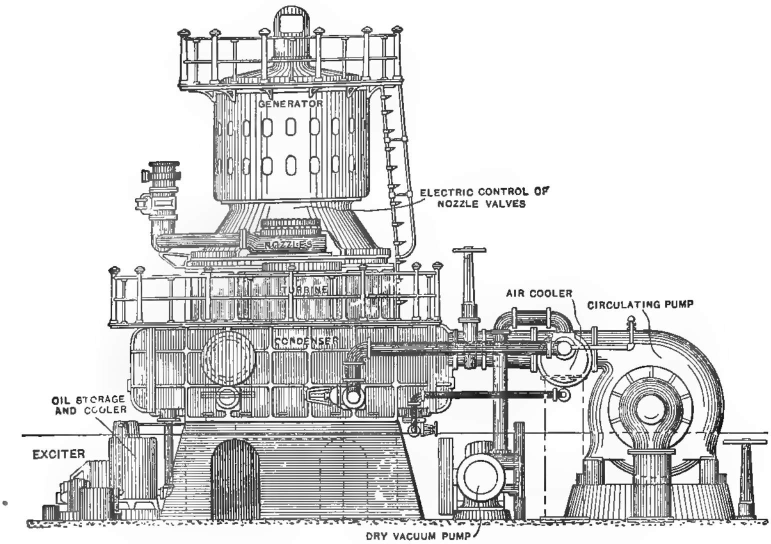


FIG. 320.—2,000-kilowatt Curtiss turbine and generator.

pump, a hot-water pump, an oil-pump, and an air-cooler. The normal speed of the turbine and generator was about 800 revolutions per minute, with an initial pressure of 175 pounds.

There are no oil-cups, and no hand-oiling is required, all lubrication

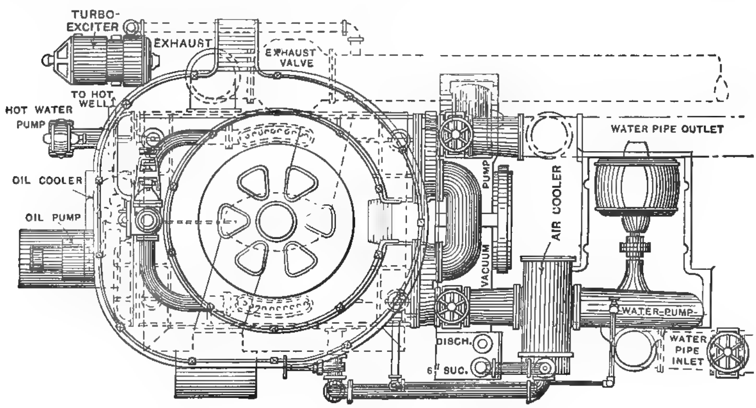


FIG. 321.—Plan of turbine-plant.

being performed by a circulating system, power for which is furnished by a steam-driven oil-pump capable of delivering $7\frac{1}{2}$ gallons of oil per minute against a pressure of 500 pounds per square inch. This pressure is maintained below the step-bearing, while a baffler allows a small quantity of oil at a low pressure to rise to a small tank on the top of the machine, whence it flows by gravity to the top and middle guide-bearings. There is a combined reservoir and cooling-tank in this oiling system of 100 gallons capacity. Absolutely no cylinder-oil is used within the turbine, nor does any oil mingle with the steam.

The most interesting of the auxiliaries is the 25-kilowatt direct-connected horizontal type turbine-driven exciter, making 3,600 revolutions per minute and furnishing direct current at 125 volts. This machine is of the single-stage type, having three rows of buckets

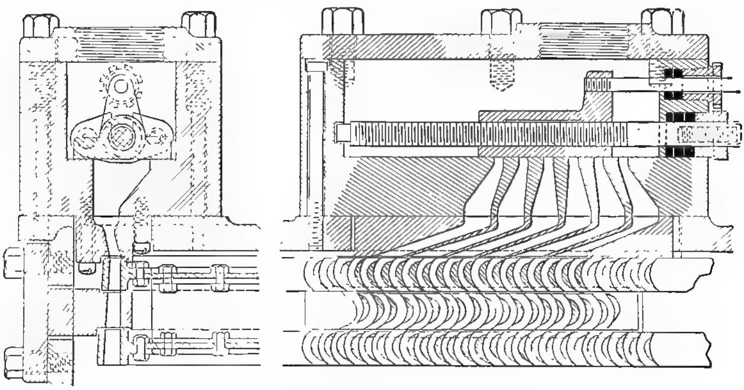


FIG. 322.—Slide-valve regulation.

on the wheel and runs non-condensing. It is governed with a throttle-governor, and maintains a practically constant speed and voltage from no load to full load. It has a forced lubricating system of its own, operated by gear from the generator end of the shaft.

In starting up either the large or small turbine, the operator merely opens wide the main steam-valve, after which both machines take entire care of themselves, whether running light or carrying a heavy, fluctuating load.

In Fig. 322 is shown in section the slide-valve system of regulating the flow of steam to the vanes of a two-stage Curtiss turbine of recent model.

In Fig. 323 is illustrated a novel arrangement by which the buckets on each disk are doubled in effect, producing eight stages of impact and reaction in a four-disk turbine. The exhaust-steam from the first disk of two sets of running buckets and intervening stationary buckets is passed, through an outside chamber with automatic valves, to the second and third disks, and thence direct through the buckets of the fourth disk. The area of the buckets throughout the system is expanded by lengthening to meet the requirement of the expanding volume of the steam.

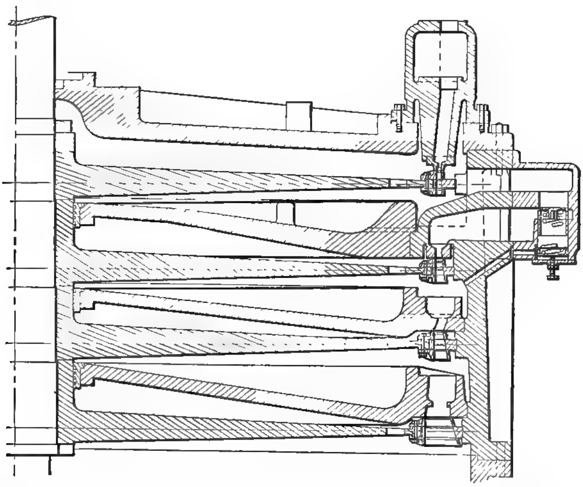


FIG. 323.—Reënforcement in four-stage turbine by double buckets on each disk.

For this construction the rim of each disk is made wide enough to receive the stationary-bucket sections and their clearance. The disk-buckets are flanged and bolted to each side of the disks. The stationary buckets are fixed in grooves in the shell.

The vertical position of the shaft in the Curtiss turbine throws great weight upon the shaft-step, which with any ordinary form of step-bearing would become an insurmountable obstacle to this position of the turbine.

In Fig. 324 is shown a section of the step and adjustments as designed for the Curtiss turbine. The bearing consists of two hard cast-iron blocks, one carried by the end of the shaft and fixed by dowels and key. The lower block is fitted to the follower, and is

supported by a powerful screw driven by a wheel, and is provided with worm-gear for adjustment.

This block is recessed to about half its diameter, and into this recess oil is forced with sufficient pressure to balance the weight of the whole revolving element. The amount of oil required is small. About 4 gallons per minute is used in the 5,000-kilowatt machine. A water circulation in the main step-block keeps all the bearings cool.

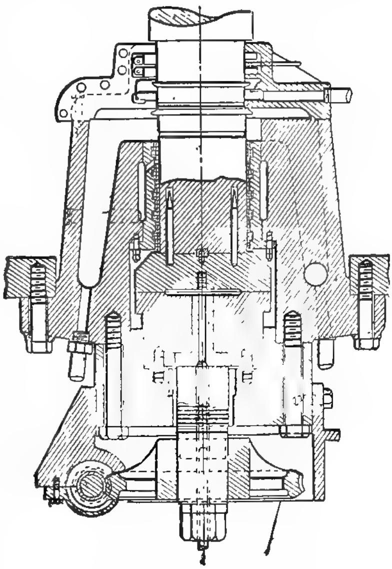


FIG. 324.—Turbine-step.

The oil, after passing between the blocks of step-bearing, wells upward and lubricates a step-bearing supported by the same casting. This whole structure is inside the base, and a packing is used between the oil-chamber and the base, so that oil or air cannot get into the vacuum-chamber. A small steam-pressure is maintained between the sections of this packing, in order that these objects may be accomplished with certainty. In many cases these same step-bearings have been operated with water instead of oil, in which case no packing is necessary, the water being allowed to pass into the base.

THE RATEAU STEAM-TURBINE

The steam-turbine designed by A. Rateau and made principally in France and Germany, is a horizontal turbine of the axial-impulse type, with the blades of the same form as those of the De Laval, Curtiss, and Parsons models, but differing in constructive detail. The bucket-orifices are enlarged progressively, as in the Parsons and Curtiss turbines, by lengthening the blades of both elements.

The revolving wheels are formed of disks of thin sheet steel, carrying cylindrical buckets on the periphery, these buckets being riveted to a band of steel welded to the disk. This gives a very light

and strong construction, maintaining its balance at all speeds. The guide-buckets are fixed in circular diaphragms, secured at the periphery in grooves cut in the interior of the turbine-case. There is thus left between the successive diaphragms a series of annular chambers in which the revolving wheels are placed. The shaft passes through bushings fitted in the diaphragms, there being but little

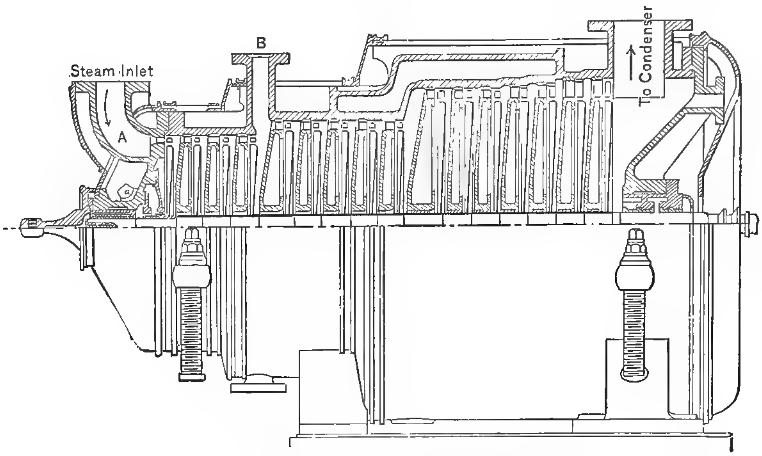


Fig. 325.—Rateau steam-turbine.

play or clearance. Between the fixed and revolving portions of the turbine, however, the clearance may readily be made as much as 5 millimetres, without injury. The main bearings of the shaft are outside the casing, a special form of stuffing-box being employed, assuring tightness against leakage.

An ordinary compensated centrifugal governor is used to regulate the speed, acting by varying the pressure of the steam delivered to the turbine. By means of a by-pass in the main steam-pipe it is possible to deliver steam of full pressure both to the entrance of the turbine at A, and to a point in the machine nearer to the condenser, this enabling a higher power than the normal amount to be produced by the machine, much in the same manner as a compound engine may be used with full-pressure steam in both high- and low-pressure cylinders.

THE ZOELLY STEAM-TURBINE

Among the many different types or models of turbines built in Europe, we illustrate in Fig. 326 a Swiss duplex turbine, constructed by Escher, Wyss & Co., of Zurich, the builders of the first Niagara water-turbines, a type since adopted by American builders. It is of the Zoelly design, and divided into two compartments, alike in detail, but one is adapted to a high- and the other to a low-pressure steam-range.

The Zurich turbine is of the multistage impulse type, the expansion of steam taking place in the passages between the wheels, and the force on the moving wheels being exerted by impact of the rapidly flowing steam-jet. On the high-pressure wheels, steam is admitted to buckets around only part of the circumference, while on the low-pressure wheels the admission is to all buckets. The high- and low-pressure ends are mounted independently on a single base, connection

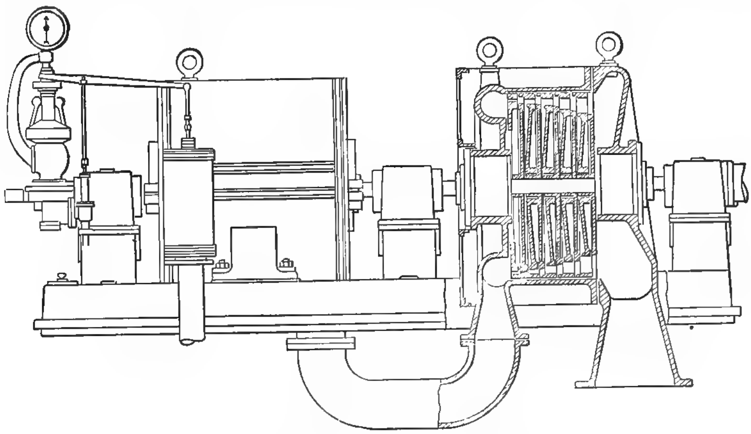


FIG. 326.—View and section of Zoelly steam-turbine.

between them being made by a pipe carried beneath the turbine. The main bearings are outside the wheel-cases, and are mounted on the base-plate independently. They are thus kept away from the heat of the steam, and are easily accessible for inspection and repairs.

The wheels are built up of open-hearth steel disks, keyed to the shaft and having on the outer rim a ring fastened in such a manner

that with the rim of the disk it forms a dovetailed groove. The buckets and spacers between them are held by tongues in this groove, and project radially from the rim of the disk. The buckets are of nickel steel highly polished to reduce friction, as are also the disks.

A point which is made by the designer is that the blades decrease in cross-section from their inner to their outer ends, so that the centrifugal force is kept very low, and the blades can safely be made much longer than if they were of uniform cross-section. Also the blades, which act as a cantilever-beam, are made strongest at the point where the bending moment due to steam-impact is greatest. The curve of the blades is such as is needed for the progressive expansion of the steam.

This special construction allows of running the wheels at a high rim-speed, thus reducing the number of stages needed in the turbine, which in turn allows of a shorter machine and of lower cost. The guide-wheels are placed between the revolving wheels and are carried from the outer casing. On account of the shape of the guide-blades, shown in Fig. 327, there is a considerable endwise pressure which is taken care of by using thick distance-pieces projecting outside the radial blades of the moving wheels and transmitting the end-thrust from the guide-disks to the outer end of the casing. The general arrangement of wheels and guide-disks is shown in the section of the low-pressure end, and the detail of construction in the larger view. The pressure on the two sides of the revolving disks is in all cases equal, so that no end-thrust is produced. The governor is of the flyball type, and operates a pilot-valve controlling the motion of a plunger, which in turn operates the main steam-valve, thus throttling the pressure of the steam. The water-pressure for operating this plunger-valve is furnished by an auxiliary motor.

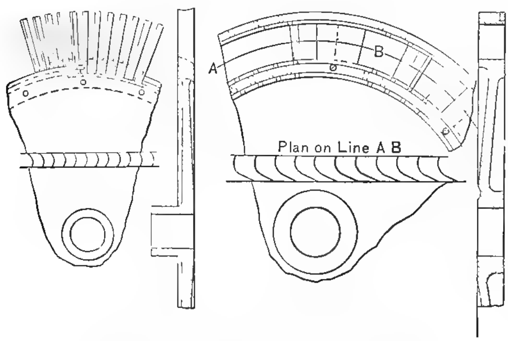


FIG. 327.—Detail of wheel and guide-disk.

We cannot give space to all the types or models of steam-turbines

in successful operation; some of them will no doubt become permanent in their special line of usefulness. They have come to stay, in harmony with the reciprocating engine, which by its long and varied trial has standardized it for every use.

THE ROTARY ENGINE

In regard to the rotary type of steam-engine we find nothing worthy of illustration or description, and from many years' experience with their performance and lasting qualities, have found nothing as yet to recommend them, being inefficient and of short life; yet a few small ones are in use. We have left them to the tender mercies of their inventors and promoters.

The Dake engine, now in use for small units of power, although called a rotary, is not of that type, but a combination of two rectangular pistons, concentric and moving at right angles to each other, of which one is pivoted to a crank and shaft centring within a rectangular case.

The contest for the survival of the fittest types of prime movers of motive power in the future will finally culminate in the success due to efficiency and durability in the various fields of their usefulness, and no one type can become universal.

Nature's elements, wind and water, with steam and confined combustion for power, are the only and real bases of our future prosperity in the production of primary power, in which each is prominent in its own sphere of usefulness.

THE STARTING AND OPERATION OF LARGE STEAM-PLANTS

Much discussion has been published in technical journals in regard to the time required for starting large steam-plants of the reciprocating and turbine types. We here give an excerpt from a published article by an engineer familiar with large steam-plants:

"So much has been written about the sensitiveness of a rotating disk to the changes of temperature and the effects of unequal expansion that it is easy to imagine difficulties in the rapid start. The possibilities of an engine with a 62-inch low-pressure cylinder in

starting practically cold and coming up to synchronous speed are well understood. A station-manager would criticise an engineer who would open his throttle as fast as he dared without wrecking his piping system and let his machine jump into her work. One turn at a time on the throttle is about all that is considered safe, and even then a close watch is kept for groaning valves and cold back bonnets.

"Every time the starting-valve is moved to increase the steam-flow the engine is allowed to take its full increment of speed, due to that particular throttle-position, before the supply-valve is moved a second time. There are ten large oil-cups, and frequently more, that must be opened and adjusted before the machine moves at all, besides whatever oiling is to be done about the air-pumps and other auxiliary apparatus.

"Most engineers would consider ten minutes as rather a fast start and fifteen minutes as a more usual starting-period, including time taken for warming up; in fact, it may not be overstating the case to say that if it were known that an engine-driven plant were to be called upon in emergency for power and it were essential that the briefest possible time were to elapse between the call and the taking of the load, one or more engines would be kept in motion all the time, turning slowly and hot all over.

"This question makes itself very prominent when the steam-station is operated as an auxiliary to a large source of high-tension power, which is itself in the construction stage and has a large over-load capacity of its own to carry, supplying all sorts of apparatus that use electric power, railway, lighting, and power circuits simultaneously.

"At such a time all sorts of accidents will happen to the high-tension water-driven plant, most of them due to the necessarily temporary character of many of the electrical connections. It takes months before an intricate system of wiring can be thoroughly relied upon, for it takes months before the temporary work of construction can be replaced.

"The station under consideration is equipped with three Curtiss turbine-driven alternators, 40-cycle, 10,000 volts, each of 1,500 kilo-watts normal capacity. During the summer months the station is operated as an auxiliary to a water-power plant, taking all sudden overloads.

"A signal has been arranged, a $\frac{3}{4}$-inch whistle, so that it can be blown instantly should the power fail. A blast of that whistle means—cut in two turbines and bring the third up to speed. The load will be heavy, and all auxiliary apparatus must be in regular operation.

"Each turbine has a surface-condenser, and there are three or four pumps to be started for each pair of turbines—one circulating-pump, one combined hot-well and feed-pump, one pressure-pump for the step-bearings, and one dry air-pump, all of which are motor-driven. The exciter is driven by a steam-engine and must be started also, for it supplies current to a portion of the auxiliary apparatus.

"The boiler-room has steam up at all times, supplying a system for manufacturing purposes other than power, and slow fires are kept in enough boilers to make steam needed for the normal load. Forced load means forced fires. The boilers have under feed-stokers, equipped with pressure-blast, and will respond quickly to a 50-per-cent.-excess call for steam. The operating force for this is about equivalent to a force for an engine-driven plant. Engineers and oilers, however, are busy about the building on construction work, installing new apparatus and taking such work as their regular occupation when the turbines are not running.

"At the sound of the whistle the water-tender starts a blower on the extra row of boilers: all blast-dampers are opened up and all stokers are allowed to feed at the maximum rate. Each fireman dumps his free ash and bars over his red fire.

"The man in charge of the coal-and-ash conveyer starts the pressure-pump for step-bearings. One of the turbine men starts the exciter which supplies current to the auxiliaries beside its field-current; a second turbine man starts the circulating-pump and then his turbine. The hot-well pump and the air-pump are started by the oiler. These movements take place simultaneously. The force is organized upon the lines that obtain in a fire-station; each man has his specific duty, and after performing it looks to see that there is nothing more for him to do. Only a few seconds elapse between starting the first pump and starting the first turbine.

"The turbine-throttle is opened as fast as an 8-inch steam-valve can be opened without endangering the steam-piping system. It is not considered advisable to open the throttle-valve as fast as a man's

strength will permit; but if nothing unusual occurs in the pipe-line, sentiment does not spare the turbine.

"One electrician attends to the switchboard and telephone. As soon as the machine approaches speed, the synchronizing system is cut in and the main switches are got ready. One and one-half minutes will do all the work here outlined, including the time taken in mustering the crew from various parts of the building, itself not a trivial matter.

"Manipulating an engine-regulator so that it shall be at a precise speed and at an exact phase relationship from some other machine, not more than $\frac{1}{1500}$ part of a second removed from it, is no matter that can be hurried, and one minute is fast time on such work. But the whole thing, phasing-in and all, has been done in two and one-half minutes, including full load on the turbine, which started from a standstill.

"This performance has been gone through a great many times, and our record-book shows that out of 43 such calls, 10 starts were made in $2\frac{1}{2}$ minutes, 18 in 3 minutes, and 15 in $3\frac{1}{2}$ minutes.

"We have taken the time in a number of instances when all the auxiliaries have been in motion and it only remained to start the turbine and phase it in on the line; the only valves to open in such cases are the throttle and one small oil-valve. The two quickest starts have been made in forty-five seconds and seventy seconds, respectively, including phasing-in. Others range between one minute ten seconds and one and one-half minutes. The two quickest starts were made on a turbine which had stood for twenty-four hours with the throttle-valve shut tight, though there was a slight leakage past the seat. After the throttle-valve is off its seat it is not more than thirty seconds before the turbine is up to speed. A cross-compound reciprocating engine of the four-valve type, 2,250 horse-power capacity, can be brought up to speed from a standstill in five minutes if it is hot all over. This five minutes is to be compared with the seventy seconds required for the similar turbine operation.

"A reciprocating engine, which is turning over slowly with the throttle-valve just off its seat or with by-pass open, and having all its oil-cups open and regulated, can be brought up to speed—say, seventy-five turns—in two and one-half minutes. This can be compared with the thirty seconds necessary for bringing the turbine up

under the same conditions; that is, about one-fifth the time necessary for bringing up the engine.

"If the engine is cold all over and has all its oil-cups shut tight and all its auxiliaries quiet, fifteen minutes is called a rapid start. Starts have been made under such conditions in twelve minutes. When we start a cold turbine, we open up the valve and let her turn, and in two minutes we are ready to bring her up to speed, and she will be at speed in two and one-half minutes, dividing the engine's time by more than four."

The points of practice here suggested from an engineer's experience in operating large steam-plants are well worthy of study and remembrance by all engineers, and their appeal is directly urged upon the student, who may profit by them in his initial trials. Their neglect has caused many wrecks in expensive installations of steam-power, resulting not only in the expense of repairs, but often the delay is the most expensive item in the wreckage cost.

CHAPTER XX

MECHANICAL REFRIGERATION-ENGINEERING

THE principal difficulties encountered in becoming a competent engineer in charge of refrigerating-machinery do not include a thorough comprehension of the fundamental principles, but are found in the arrangement of the piping and valves of which the greater part of the system consists. It requires considerable practice to learn what to do and when to do it, and to be able to note the little changes and minor adjustments which affect the economical production of low temperatures.

Refrigeration in principle can be learned without special effort, but the proper manipulation of the various valves and knowing what results should be obtained at the various points in the system require the assistance of an experienced person. Even practice may be obtained without special instruction; but experimenting with ammonia is usually found to be very different from experimenting with steam or water under like pressures, and it is liable to lead to accidents and unnecessary expense.

The more noticeable effect upon the general demeanor of the successful refrigerating-engineer, due to a thorough knowledge of the peculiarities and requirements of ammonia, is that of making him careful and thorough in every detail of his work. Makeshifts in a refrigerating-plant cannot be tolerated. Whatever is done must be done thoroughly, in order to avoid increasing annoyance, if not unnecessary expense and actual danger. An engineer accustomed to operating a refrigerating-plant will usually be found a careful person in any plant. Beside this he will have broadened his knowledge of the compression and expansion of gases and of the generation and removal of heat and its effect upon various substances and liquids; in fact, he will have taken another step forward toward the mastery of the various sciences underlying and connected with steam-engineering.

Engineers can scarcely expect to escape refrigerating-machinery, no matter where they go. Small dairies and cheese-factories located in farming districts, abattoirs on country roads; in fact, any establishment large enough to warrant the use of a steam-boiler and in which low temperature is required for the preservation of the product is likely to, and oftentimes does, contain a refrigerating-apparatus of one kind or another.

Thus mechanical refrigeration has become one of the branches of steam-engineering—one that has made a considerable demand upon the ingenuity and resourcefulness of the engineer, and has become so important a part of the engineer's education that the time is not far distant when the steam-engineer will not be considered thoroughly competent without a working knowledge of both the compression and the absorption system.

ANHYDROUS AMMONIA

Ammonia is a gas composed of 82.35 per cent. of nitrogen and 17.65 per cent. of hydrogen. It is very much lighter than air, its specific gravity being 0.589. It is characterized by a pungent, suffocating odor, and by its high solubility in water, one volume of water at 32° F. absorbing 1,050 volumes of gas. This solution of the gas in water is what is commonly known as aqua ammonia, and it rather confuses the situation, because the water-solution of gas is used in the absorption system of refrigeration, while the ammonia used in the direct expansion and compression system is an entirely different product. The product known as anhydrous ammonia is the gas itself liquefied by intense pressure. It has absolutely no water content, and is strictly analogous to liquid air, but liquid air consists of two distinct substances, each one a gas liquefied by intense pressure—that is to say, the nitrogen and hydrogen in anhydrous ammonia are chemically combined to a single substance, while in liquid air the oxygen and nitrogen composing the air are not chemically combined.

Anhydrous ammonia boils at a fixed temperature of 28.6° F. below zero. One pound of liquid ammonia at 32° F. would occupy 21 cubic feet when evaporated to a gas at atmospheric pressure, and the vaporization of a pound requires 555.5 British thermal units. It

is a colorless and very mobile liquid. It has a specific gravity of 0.613 compared with water at 60° F. A cubic foot of anhydrous ammonia weighs 42.1 pounds.

The usual method of detecting impurities is by evaporation of a measured amount of the substance. The residue remains in the bottom of the tube, and can be either weighed or measured. Not many years ago it was quite common to have ammonia containing as high as 15 per cent. of impurities; to-day commercial ammonia rarely contains 0.5 per cent. of impurities. It is stated that Armour's anhydrous ammonia does not exceed 0.1 per cent. of impurities, and usually but the merest trace. Impurities amounting to 2 per cent. are detrimental, as they affect the refrigerating value of anhydrous ammonia. Impurities of 0.1 per cent. or less can be disregarded. It is true these impurities accumulate in the system, but when the amount is but 0.1 per cent., this has little effect on the action of ammonia in the refrigerating-plant, and its accumulation is very slow.

Without going into a discussion of the availability of the several different gases for refrigerating purposes, it may be said that ammonia combines the required characteristics and therefore is found to be most suitable; for when we consider the pressures at which other gases can be made to liquefy when at ordinary temperatures and the amount of cold water that otherwise would be required, together with the important item of safety or the absence of dangerous qualities, it is easily understood why ammonia is best adapted to the purpose. Therefore we will consider the ammonia compression system of refrigeration from the standpoint of the engineer in charge of the machinery and whose success in handling it will be directly in proportion to his knowledge of the principles involved, together with the details of the machinery and the care bestowed upon it.

When ammonia is received ready for the system it is in the liquid state enclosed in steel drums, which are only partly filled, leaving space enough for expansion so as to prevent an explosion of the drums. Ammonia-drums have exploded, but always under conditions of overheating, for in general, with proper care, there is no danger. When liquid ammonia evaporates into gas under a lower pressure, heat must be added to supply the latent heat of the gas corresponding to that pressure and temperature. The latent heat of the gas and

many other points regarding pressures and temperatures may be found from the following table, which gives the more important properties of ammonia.

TABLE XL.—PROPERTIES OF AMMONIA.

Gauge-pressure, pounds per square inch.	Absolute pressure, pounds per square inch.	Temperature, degrees F.	Absolute temperature, degrees F.	Latent heat of evaporation in thermal units.	Volume of 1 pound vapor in cubic feet.	Weight of 1 cubic foot of vapor in pounds.	Volume of 1 pound of liquid in cubic feet.	Weight of 1 cubic foot of liquid in pounds.
−4.01	10.69	−40	420.66	579.97	24.38	.0410	.0234	42.589
−2.39	12.31	−35	425.66	576.68	21.32	.0469	.0236	42.337
−0.57	14.13	−30	430.66	573.69	18.69	.0535	.0237	42.123
1.47	16.17	−25	435.66	570.68	16.44	.0608	.0238	41.858
3.75	18.45	−20	440.66	567.67	14.51	.0690	.0240	41.615
6.29	20.99	−15	445.66	549.35	7.23	.1383	.0243	41.375
9.16	23.80	−16	450.66	546.26	9.49	.1541	.0250	41.135
12.22	26.92	− 5	455.66	543.15	5.84	.1711	.0252	40.895
15.67	30.37	0	460.66	540.03	5.27	.1897	.0253	40.655
19.46	34.16	5	465.66	536.91	4.76	.2099	.0255	40.415
23.64	38.34	10	470.66	549.35	7.23	.1373	.0249	40.160
28.24	42.94	15	475.66	546.26	6.49	.1541	.0250	39.920
33.25	47.95	20	480.66	543.15	5.84	.1711	.0252	39.682
38.73	53.43	25	485.66	540.03	5.27	.1897	.0253	39.432
44.72	59.42	30	490.66	536.91	4.76	.2099	.0255	39.200
51.22	65.92	35	495.66	533.78	4.31	.2318	.0256	38.950
58.29	72.99	40	500.66	530.63	3.91	.2554	.0258	38.700
65.96	80.66	45	505.66	527.47	3.56	.2809	.0260	38.480
74.26	88.96	50	510.66	524.30	3.24	.3084	.0261	38.230
83.22	97.92	55	515.66	521.12	2.96	.3380	.0263	37.980
92.89	107.59	60	520.66	517.93	2.70	.3697	.0265	37.736
163.37	118.03	65	525.66	514.73	2.48	.4039	.0266	37.481
114.49	129.19	70	530.66	511.52	2.27	.4401	.0268	37.230
126.52	141.22	75	535.66	508.29	2.09	.4791	.0270	36.995
139.40	154.10	80	540.66	505.05	1.92	.5205	.0272	36.751
153.18	167.88	85	545.66	501.81	1.77	.5649	.0273	36.509
167.92	182.62	90	550.66	498.55	1.64	.6120	.0275	36.258
183.65	198.35	95	555.66	495.29	1.51	.6022	.0277	36.023
200.42	215.12	100	560.66	492.01	1.39	.7153	.0279	35.778
218.28	232.98	105	565.66	488.72	1.289	.7757	.0281	....
237.27	251.97	110	570.66	485.62	1.203	.8312	.0283	....
259.70	272.14	115	575.66	482.41	1.121	.8912	.0285	....
275.79	293.49	120	580.66	478.79	1.061	.9608	.0287	....
301.46	316.16	125	585.66	475.40	.9699	1.0310	.0289	....
325.72	310.42	130	590.66	472.11	.9051	1.1048	.0291	....

The ammonia-compressor is a compression-pump, and may be considered as such in every sense of the word, but it must be a better pump than those which are more common in the steam-plant. The object of this pump is to take ammonia gas from the refrigerating portion of the system, and compress it to a considerable pressure and discharge the compressed gas into the condenser. The latter is a series of pipes over which water is kept flowing for cooling, and, finally, for the liquefaction of the gas. The ammonia, on being liquefied in the condenser, passes on to the ammonia-receiver, which is a vessel of any convenient size and shape adapted to hold a suitable quantity of the liquid and from which a considerable quantity may be withdrawn continuously and evaporated in the refrigerating-pipes. In the latter, sufficient heat is absorbed to supply the latent heat required by the gas, after which the gas returns to the compressor.

With the foregoing brief description of the essentials of the system, the engineer may proceed to start the compressor for continuous work. The compressor may be driven by a steam-engine or electric motor, and in some cases water-power is used for this purpose. When shutting down their engines or stopping the compressor for any reason, some engineers leave the discharge- and suction-valves in the ammonia system open, as they were during the time the machine was running, while others often close the suction- and discharge-valves and sometimes forget to open them before trying to start. Do not make this mistake, for many accidents have happened because the discharge-valve was closed when the compressor was started.

There is not as much danger in leaving the suction-valve closed, for the worst that could happen would be to create a high vacuum on the suction side. If the discharge-valve happens to be closed, each stroke of the compressor will add to the pressure in the discharge-pipe, which will soon run up to a dangerous point. The careful engineer will keep his eye on the pressure-gauge while starting the compressor and until he is assured that the compressor is running at the proper speed and that there is a free escape for the ammonia. Usually there are no cylinder-cocks to be opened previous to the starting of the compressor and none to be closed, except on the steam end. There is little chance therefore for the engineer to make a mistake, provided the pressure-gauge shows that the pressure is within the proper limits.

Forward or compressor pressure required is determined in nearly all cases by the temperature and amount of the condensing water. Further information on this point may be obtained by examining the temperatures and consequent pressures as given in the table, where it will be found that ammonia under a pressure of 200 pounds to the square inch has a temperature of about 100° F., while the cooling water may have a temperature of only 50°. The boiling-point of ammonia under 200 pounds pressure with a temperature of 100°, shows that the temperature of the compressed gas must be reduced below 100° in order to be able to liquefy it.

There is a difference of a few degrees between the temperature of the gas or liquid inside the pipe and the temperature that will be obtained on the outside—that is, where the ammonia-pipes in the condenser are clean and free from scale, mud, and slime. The difference of temperature inside and outside the pipes will range from 5 to 8° under usual working conditions. With unfavorable conditions the difference in temperature may be almost any amount. A difference of temperature will also be found in the refrigerating-pipes, and this will be found to be about the same number of degrees under the same conditions.

With a gas-pressure in the condenser of 200 pounds and with water at 50° flowing over the condenser-pipes, the best conditions will be obtained when the temperature of the water leaving the condenser is within a few degrees of the temperature of the ammonia inside the pipes. The temperature due to the pressure of the ammonia gas in the condenser is always a few degrees higher than that due to the boiling-point of liquid ammonia under the pressure carried. This difference of temperature is due to the superheating of the ammonia when compressed in the compressor, which will vary according to the conditions of operation and the kind of compressor used.

Suction-pressure on the system is nearly always a few pounds above that of the atmosphere, although it is sometimes found necessary to reduce this pressure below that of the atmosphere for special kinds of work. For economical results the less difference between the condenser- and suction-pressures the less power will be required to operate the system. For a given amount of refrigeration the suction-pressure usually is regulated by the lowest temperature required in all parts of the system, provided the proper amount of

piping is employed. The engineer must make sufficient allowance for the loss of temperature by transmission through the pipes, which, as previously mentioned, may amount to from 5 to 8°, but sometimes may reach a much higher figure. Reference to the table of pressures and corresponding temperatures will indicate the proper suction-pressure for the lowest temperature it may be required to reach.

When considering the principles of operation of a mechanical refrigerating-plant, we shall see that the effects produced are all due to a simple exchange of heat, for when we compress the gas we squeeze out, so to speak, a certain portion of the latent heat, and the gas being under pressure, its latent heat is less than when under a lower pressure. The latent heat taken up by the water requires a much

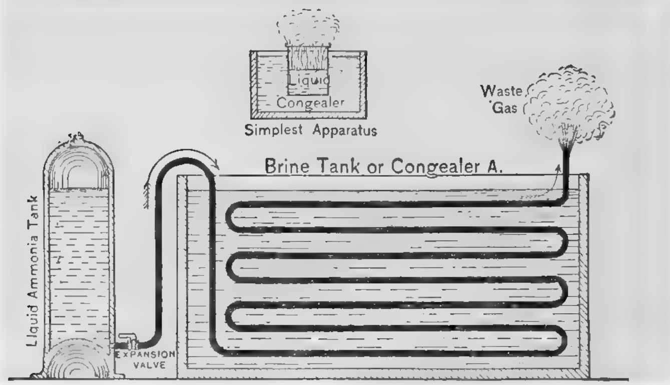


FIG. 328.—Diagram illustrating the principles of refrigeration by ammonia.

larger quantity of water for condensing purposes than that due merely to the difference between the temperature of the compressed gas and that of the liquid ammonia under this pressure.

Superheating of the gas is due to the latent heat set free by the greater density of the compressed gas, and it is the removal of this latent heat which gives the liquid greater capacity for absorbing heat, and as heat-absorption is the object to be attained the ammonia should not be fed to the refrigerating portion any faster than it can absorb heat. Furthermore, when it reënters the compressor it should be all gas, except in cases where a small amount of liquid is permitted to enter the compressor for purposes which will be explained later.

One way in which the proper pressure on the suction side may be determined is by frost covering the suction-pipes. As long as there is frost on the pipes it shows that there is still unevaporated ammonia in the pipe. Some engineers do not appear to understand clearly whether it is the ammonia gas or the liquid ammonia that absorbs heat. A simple test will determine this point in a way that will render clear to the average person just what takes place in the pipe when partly filled with ammonia. It will also demonstrate the temperature produced on the outside of the pipe. Ammonia gas has little effect in absorbing heat, because the gas is already supplied with nearly the full quantity of latent heat required to keep it in the gaseous condition. This point can be made clear by taking a test-flask or a common tumbler partly filled with liquid ammonia and exposing it to the atmosphere. As heat is supplied from the flask and the surrounding atmosphere, the ammonia will begin to boil to a noticeable degree, which, however, will continue for a moment only, for the contents of the flask soon become cooled to such an extent that a coating of frost is produced on the surface by the condensation of moisture from the atmosphere.

Frost thus produced will increase in thickness until a layer is obtained through which heat can pass less readily; then the boiling of the ammonia will be greatly reduced, and only small bubbles of gas will be seen rising through the liquid and given off at the surface. The level of the liquid ammonia in the flask is marked by the frost-line on the outside, no frost whatever appearing at a greater height than the level, except perhaps for $\frac{1}{8}$ inch or so where the warmer gas is in contact with the glass. Air on the outside and the gaseous ammonia on the inside cause the frost to melt and form a thin ring of ice or a mixture of ice and water at the line where gas and liquid meet. This experiment should prove conclusively that it is the liquid ammonia that does the work of absorbing heat under the conditions noted, in which case ammonia gas under pressure, the conditions may be considered as being the same as in the glass, for the latent heat in the ammonia gas at the suction-pressure has been fully supplied, and the gas has no further capacity for absorbing heat. Therefore the liquid ammonia only is available for that purpose. Thus we can readily understand that ammonia passing through the pipes in the refrigerating system remains partly in the liquid state as long as there

is frost on the pipes, and that the point where the liquid ceases to exist as such is marked by the absence of frost.

The amount of liquid ammonia to be supplied to the refrigerating system is thus determined by the coating of frost on the pipes, since the presence of frost is a sure indication that there is some liquid

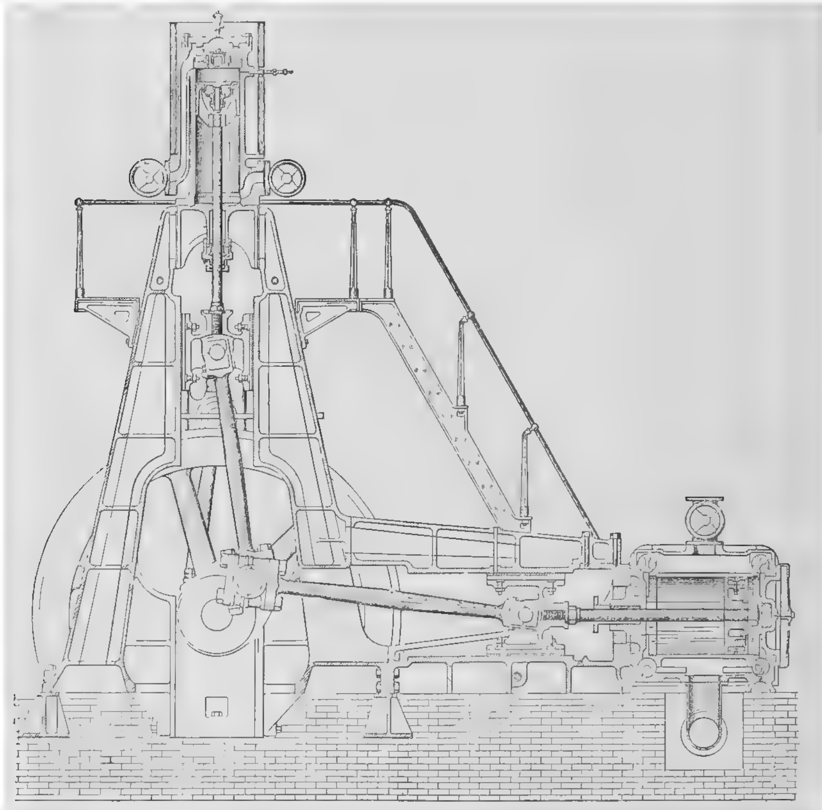


FIG. 329.—Ammonia-compressor.

ammonia in the pipe at that point, while the absence of frost indicates that the temperature of the pipes cannot be below the freezing-point. In some systems frost is carried back to the compressor; in other systems frosty pipes are only carried inside the cooling-rooms. It may be said, regarding the presence of frost on the pipes and the absence of it, that so long as we are dealing with ammonia gas we have only the specific heat of the gas to aid us in obtaining the cooling

effect, but in the transformation from the liquid to the gaseous state we have not only the specific heat of the liquid but the latent heat of the ammonia. The specific heat would not pay for the work required in compression, for the latent heat is what is most important, and it may be considered that it is all that is available in evaporation of liquid ammonia.

It is well known by all engineers that the compression of air or gas develops a large amount of heat, while the expansion of a gas will absorb heat, thus producing a lower temperature of the surroundings. It is also well known that the evaporation of a liquid, which is thus transformed into a gas, will absorb heat, and that the amount of heat thus absorbed will be equivalent to the latent heat of the gas at the pressure under which it is generated.

Fig. 329 illustrates this principle for operating a compressor. A simple non-condensing automatic engine drives a crank-shaft in the usual way, but a vertical connecting-rod is driven by the same crank-pin, and this gives motion to a vertical compressor, as shown. A heavy balance-wheel on the same shaft provides steady motion for the moving parts by absorbing power during the first part of each stroke and giving it out during the latter part. The illustration shows a

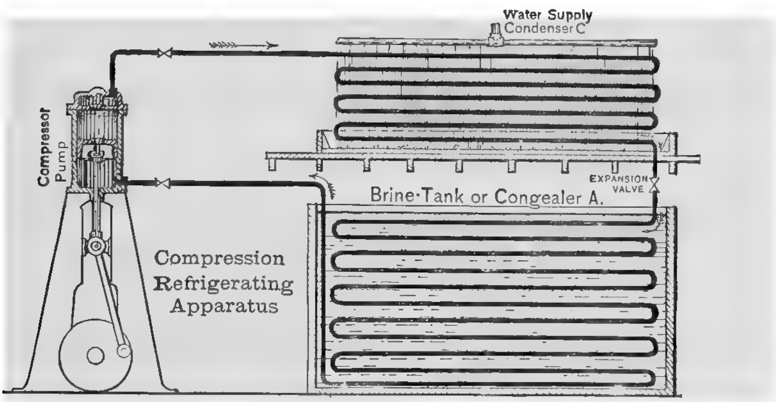


FIG. 330.—Three stages of refrigeration.

duplex compressor. One cylinder is located just over the engine-crank, while another crank and cylinder are placed on the other end of the shaft.

In Fig. 330 are illustrated the three principal phases or stages in

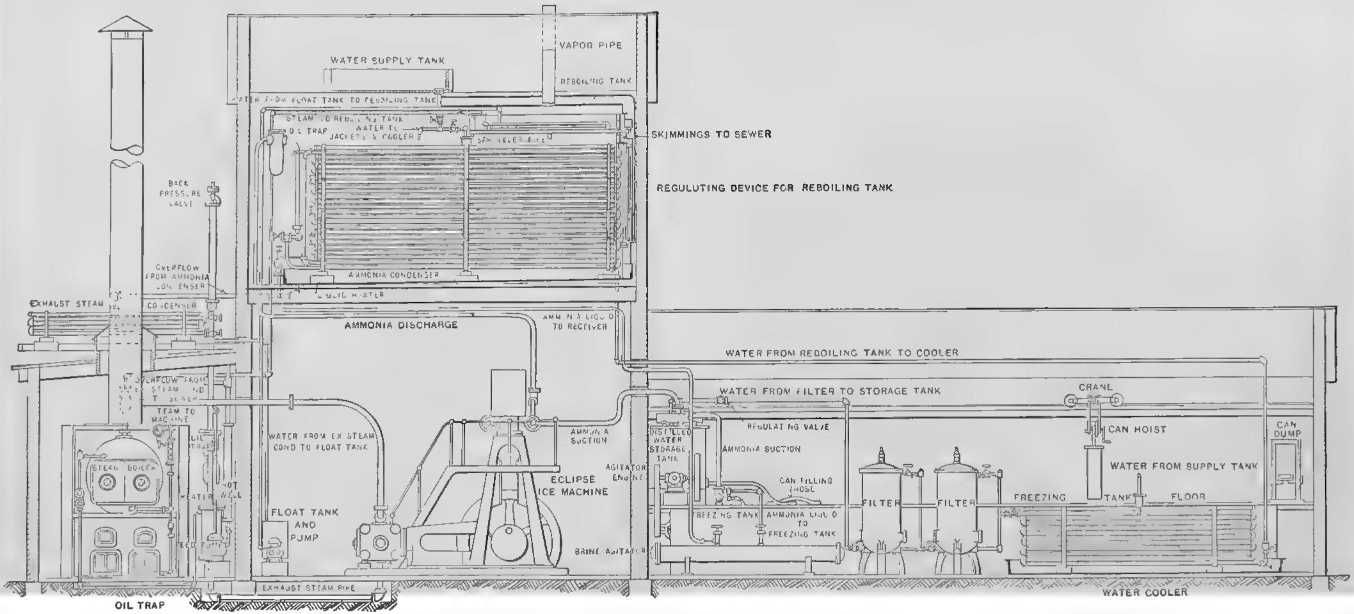


FIG. 331.—Complete ice-making plant of the Frick Company illustrating the various stages in the operation of refrigeration.

the operation of refrigeration by the ammonia compression system, although in a complete plant there are many adjuncts for special service, such as oil-separators, receiving-tanks, filters, and brine-agitators, which are shown in the full-page cut, Fig. 331.

In the simple round of ammonia-circulation, it starts from the compressor under a high pressure and temperature, passing to a cooling-coil, which is the condenser, where, by means of a cold-water sprinkler, the gas is cooled to 45° or 50° F. At that temperature, under the high pressure the gas is condensed to its liquid state and passes to a storage-tank, or may be throttled by a valve to maintain a constant pressure on the liquid, and, by allowing of control in its issue to the refrigerating-coil, and by its reëvaporation therein under a low pressure, to absorb the heat of the brine or air in a cold chamber that is required for vaporizing the liquid ammonia within the coils.

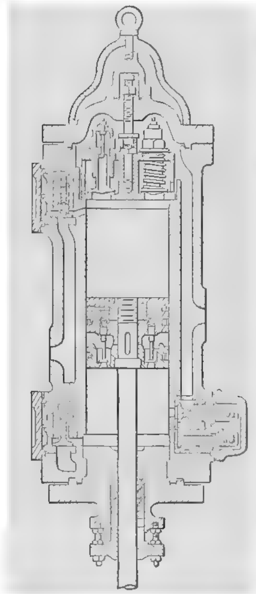


FIG. 332.—De La Vergne vertical ammonia-cylinder.

Fig. 332 illustrates the De La Vergne standard double-acting vertical compressor, the operation of which may be explained as follows: Suppose that the piston is ascending with a charge of gas above it. As the space holding this gas becomes less its pressure rises until it is high enough to overcome that carried on the discharge-pipe. There are two discharge-valves in the upper part of this machine, a full view of one and a section of the other being shown. These rise and let the compressed gas out through the right-hand passage to the condenser. At the same time gas under light pressure is drawn in through the lower suction-valve, at the left hand, until the space below the piston is filled with it. As the downward stroke is made this gas is compressed until the pressure is high enough to force it out, through the two lower discharge-valves at the right, to the condenser.

There is a hollow space in the piston, covered by two valves opening upward. When the piston has nearly reached the end of its downward stroke and the lower valve is closed by the piston,

the pressure is sufficient to raise the valves and discharge the gas, through the higher one of the right-hand discharge-valves, to the condenser.

Fig. 333 shows the sectional details of the single-acting compressor of the Frick Company. The gas at suction-pressure enters below the piston from the right-hand valve, passing through a light spring-balanced valve in the piston, is compressed above the piston, and is discharged, through the lifting of the large spring-held valve, at the top of the piston, and to the condenser.

The principal feature in this design is the safety-head or discharge-valve, which allows the piston to touch it at each stroke, thus eliminating all clearance and adding its effect to the efficiency of the compressor. The apparent striking of the discharge-valve at the moment of the passage of the crank-centre, or even slightly before it, can do no damage, as the valve is lifted at that moment and falls gently upon the piston-head during the last of the discharge.

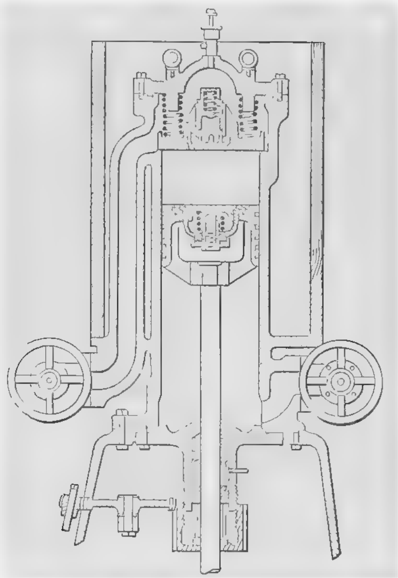


FIG. 333.—Frick Company ammonia-cylinder.

Fig. 334 illustrates a section of the De La Vergne horizontal double-acting ammonia-compressor cylinder. It has some features of safety not covered in many gas- or air-compressors. This cylinder has as small clearance as possible, in view of which it will be seen that the valves must be as near the piston as practical, in order to make the clearance small and produce an economical machine. The suction-valves are above the cylinder in this case and the discharge-valves below it; consequently if any liquid finds its way into the cylinder it is well drained out. None of the valves can drop into the cylinder in case the springs break, which is an important consideration.

In all double-acting machines the piston-rod stuffing-box is subjected to the full compression-pressure, which may be over 200 pounds,

and as this gas is of a penetrating nature, especially when under such high pressure, it is sometimes difficult to keep a stuffing-box tight

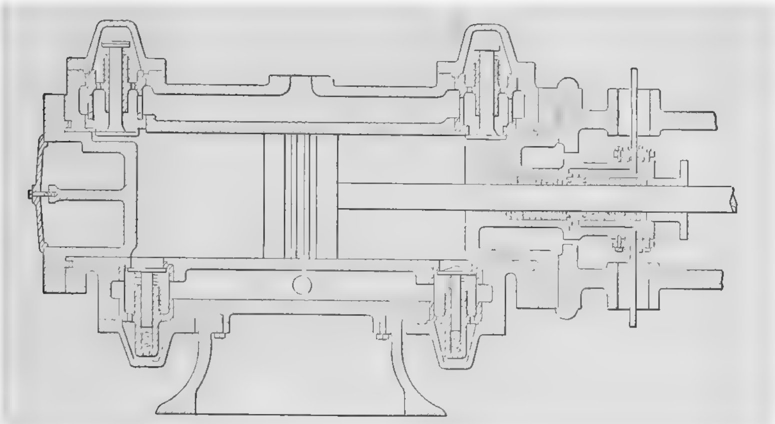


FIG. 334.—De La Vergne horizontal ammonia-cylinder.

without excessive friction. This is accomplished here by the use of two sets of packing-rings, and between them there is a device for oiling the rod, which is plainly shown.

SURFACE- AND DOUBLE-PIPE CONDENSERS

The Linde surface-condensers (a portion of which is shown in Fig. 335) are built in such a way that the flanges at each end of the straight pipes, when screwed together, form a hollow column, which, by means of special flanges, is divided into different compartments. The warm ammonia gas, when discharged into the top of one of the columns, is divided, so as to reduce the velocity, and then passed through three pipes. At the other end the three pipes join, and the gas, after being mixed, is again divided into six or more pipes, so as to still further reduce the velocity and give it time to become thoroughly cooled. This action is repeated until the ammonia is delivered at the bottom of the condenser in liquid form. By reducing the velocity the friction also is reduced, and condensation is effected in a shorter time. The pressure also is considerably reduced, which in some cases amounts to 25 pounds.

From the lower part of the condenser the liquid ammonia is drawn

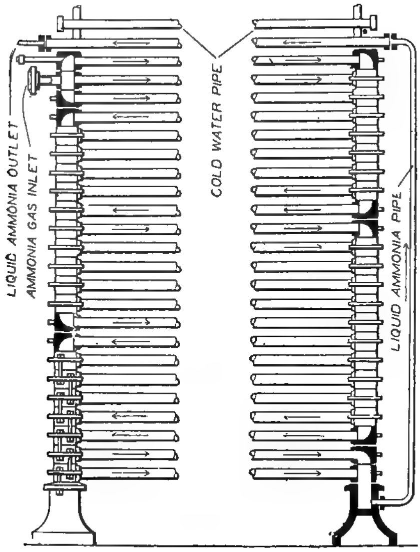


FIG. 335.—Surface-condenser.

off and passed through a separate upper pipe, where it is brought in contact with the coldest water and cooled down as near as possible to the temperature of the water.

The construction of the double-pipe condenser, Fig. 336, is such that one pipe is placed within the other, which pipes, at each end, are connected by special bends, so as to make two separate zigzag sections of inner and outer pipes.

Ammonia gas enters at the top and is forced downward through the large outer pipe, while the water enters at the bottom and is forced upward through the smaller inner pipe. As the water does not come in contact with the atmosphere, after having cooled the gas it can be used for other purposes. As no water runs over the outside pipes, no tank is required to collect the condensing water, and therefore the condenser can be placed in any room, provided the temperature of the room is not too high. On account of placing one pipe within the other the space allowed for the ammonia gas is small, and consequently the amount of gas surrounding the water-pipes is also small, so that heat is quickly extracted. This is an

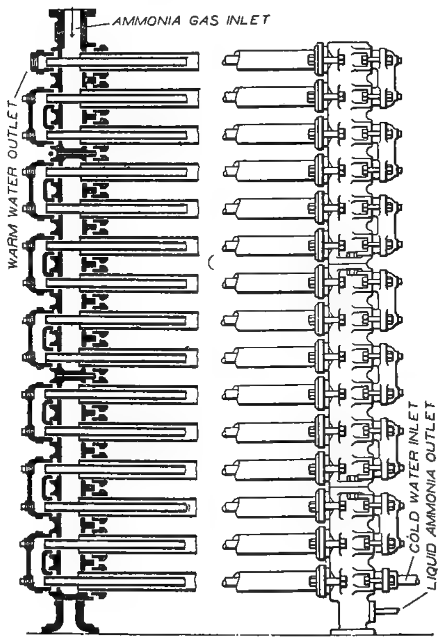


FIG. 336.—Double-pipe ammonia-condenser.

advantage, since the quicker ammonia gas is cooled and liquefied, the lower will be the pressure, and less power is required to drive the compressor. As the coldest water enters at the lower end of the coil-section, where the liquid ammonia collects, a thin layer of liquid ammonia is quickly reduced to the temperature of the coldest water. The surface-condenser has the advantage that the pipes are always open for inspection, and can be cleaned and painted when necessary, and always kept in good condition. With a double-pipe condenser, where the water-pipe is inside, they cannot be so readily examined, but special provisions are made for cleaning the pipes. In certain cases where the required quantity of cooling water is limited, or where water is metered and must be paid for, a condenser of special design is built with the object of saving water. With these condensers, which are called evaporative condensers, the quantity of cooling water is reduced to one-tenth the quantity required for ordinary condensers. The latter condensers, on account of their special construction, are somewhat more expensive in first cost, but where water is scarce or has to be paid for they soon pay for themselves.

THE DIAGRAM OF AMMONIA-COMPRESSION

The specific heat of anhydrous ammonia is about the same as that of water, or, more exactly, 1.096 at 0° F., and decreases with the rise in temperature at the rate of .0012 per degree F.

The latent heat of vaporization at $-40°$ F. is 579.6 thermal units per pound, sustaining a pressure of 10.7 pounds per square inch. Its latent heat decreases gradually with increasing temperature and pressure, and at 100° F. is 491.5 thermal units per pound, and sustaining a pressure of 215 pounds per square inch.

The compression of its vapor follows the adiabatic law of gases and vapors, subject to the influence of the walls of the cylinder in absorbing the heat of compression.

Fig. 337 shows a diagram of the compression-lines for ammonia vapor between the return-pressure of 20 pounds and discharge-pressure of 150 pounds per square inch. The adiabatic line is represented by the logarithmic exponent of the P V equation, which is $^{1.297}$, or $^{1.3}$ as generally expressed; the equation in which the $P V^{1.3} = P_1 V_1^{1.3}$ represents the integration of the curve.

It will be seen that the absorption of heat by the cylinder-walls drops the line of compression below the adiabatic line, and thus

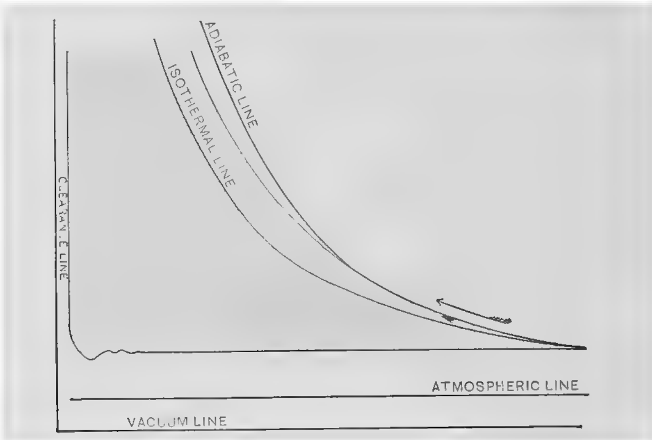


FIG. 337.—Diagram of ammonia-compression.

contributes to the efficiency of the compressor, and also shows the volume of delivery between the observed temperatures.

POINTERS ON THE OPERATION OF AMMONIA COMPRESSION SYSTEMS

It is not the intention of the author to go very deeply into the theory of mechanical refrigeration, as the practical end of the business is where trouble is generally found; but it is absolutely necessary for the engineer to have the right foundation on which to base his practice, and to assist in this a few definitions and rules will be given.

Mechanical refrigeration is brought about by an exchange of heat between two bodies; and it is well to remember that whether heat is sensible or latent it is never destroyed, but simply removed or absorbed by another body whose temperature is lower than that of the body from which heat is taken. Heat that manifests itself by means of the sense of feeling or by the aid of a thermometer is called sensible heat. In changing a solid into a liquid, or a liquid into a gas, a certain amount of heat is required to effect the transformation, and this is called the latent heat.

The process of refrigeration by the compression system is divided

into three stages, compression, condensation, and expansion. The ammonia gas is first drawn into the compressor and compressed to approximately 150 pounds per square inch. During compression the latent heat of the gas, which in this case is the amount of heat absorbed in its transformation from a liquid into a gas, is given up in form of active or sensible heat. Some compressors have a water-jacket cylinder to prevent this heat from doing damage by destroying lubrication, but as the jacket has only a local effect, it is sometimes found necessary to inject oil in large quantities, and this generally causes trouble by passing the oil-trap in the form of a vapor and coating the condensing system. Some compressors are so arranged that a small portion of the liquid ammonia is carried back to the machine and converted into gas during the compression period, and that the latent heat thus absorbed assists in keeping the temperature of compression down to a point where water-jackets are not necessary. If the temperature can be kept down to about 120 to 130° F. no trouble will be experienced from that source.

When gas at 150 pounds pressure is forced into the condenser, the cooling water running over the pipes absorbs the active or sensible heat developed during compression, thus removing the heat which was necessary to keep the ammonia in a gaseous state, and again transforming it into a liquid at a temperature approximating that of the condensing water, but at the pressure existing in the condenser. The liquid ammonia is admitted to the expansion-coils through a regulating- or expansion-valve in such quantities as are necessary for the work on hand. In these coils, owing to the lower pressure maintained, the liquid ammonia again expands into a gas, and during this transformation absorbs practically the same amount of heat from surrounding objects that it gave up to the cold water in the condenser.

The economical operation of a plant of this kind requires two things, viz., pure ammonia, as the boiling-point of ammonia varies directly in relation to the purity, and keeping the system in such a condition that ammonia will not be lost as a result of leaks. This trouble is one frequently met with. The compressor runs smoothly and everything seems to be as it should, but perhaps the proper results are not being attained in the pipe-lines. Perhaps direct-expansion piping does not frost up as it should and brine tempera-

tures are too long falling. The usual trouble is the lack of liquid ammonia in the system, or some obstruction at the expansion-valve. If there is sufficient ammonia the gas will be running heavy enough to make a very distinct clicking at the valves in the compressor, while with a lighter gas caused by a lack of liquid these valves will be almost if not entirely noiseless. If the trouble is at the expansion-valve it is generally easy to detect it by opening and closing the valve several turns and listening to the passage of the ammonia, for if the valve is at fault the sound will remain the same at all positions.

Having a machine too small for the work will also make a poor showing at the expansion-coils, but if there is plenty of ammonia in the system, trouble from this cause will also be accompanied by a high back pressure, as the ammonia expands to a gas faster than the machine can take care of the gas, and in consequence the back pressure will build up until the extra pressure in the expansion-coils is sufficient to retard the inflow of liquid ammonia and the consequent evaporation. If the expansion-valve is found to be passing gas, or if the temperature of the pipe between the liquid-receiver and the expansion-valve is found to be much below that of the condensing water, the engineer will be safe in assuming that the supply of ammonia in the system is too small. The condenser-coils should be kept free from permanent gases by the use of a gas- or purge-valve located at the top of the coils, and they should be kept as clean as possible at all times so that the entire benefit of the water may be derived.

In looking for leaks in the system, they may be quite easily located by making long sulphur matches out of pine splinters by dipping them in melted sulphur, and, after lighting, holding one of them close to and around the point thought to be leaking. If the leak is there, the sulphur fumes and the ammonia fumes will combine and form a dense white vapor. This is also a good point to remember where direct expansion is used in the cold-storage rooms, as in case of a break or a severe leak the ammonia gas can be neutralized by this method, merely placing a pan of burning sulphur inside the room. In this way work can be started much sooner than would be possible, unless there is some good means of ventilation, which as a general thing is not provided. A good-sized stream of water from a hose directed on a

serious break in an ammonia-pipe will sometimes enable the engineer to get to the stop-valve and close it before the whole charge of ammonia is lost. A common source of small leaks is the piston-rod stuffing-box; and the engineer should use great care in packing this box, because a leak at this point is both costly and disagreeable. The packing should fit the stuffing-box snugly, and be cut to lengths so that the ends will meet but not overlap. This packing should be tight enough to require tapping into place with a wooden packing-stick and small hammer.

Coils of ammonia-condensers usually are vertical pipes connected with return-bends. Should a leak develop near the centre of the coil, the quicker remedy is to cut the nearest return-bend with a hack-saw and remove the two pieces, after which the leaky pipe may be attended to properly and the joint made by using a return-bend made in two or three parts and clamped together with bolts. While on the subject of pipe-joints for holding ammonia, several kinds that give good service may be mentioned. A joint may be made by tinning the fitting and the end of the pipe and heating them hot enough to make a sweat-joint when screwed together. If an annular space is made in the fitting about two threads deep, and if after making the fitting up tight, this space is filled with solder and wiped off, it makes an excellent joint, but it is slow, costly work and requires careful handling. A common way is to clean the threads with naphtha or gasolene, and then paint them with a pigment made of glycerine and litharge. This will harden in a short time, and if carefully put up will give excellent results.

In making pure crystal can ice perhaps the greatest difficulty the engineer will encounter will be to keep it clear and free from cores. Absolute cleanliness is the greatest help toward attaining this end. The red core is caused by iron oxide from the steam-condenser coils getting past the filters, which is something that can be prevented if proper care is taken. The boilers should be kept clean along with the rest of the plant, as they are the source of the distilled water, and some little water is apt to be carried over with the steam. This may not be much, but if the boilers are dirty it will often show in the ice. A leak in the steam-condenser will often bring in enough foreign matter to cause discoloration in the ice.

If the temperatures in a cold-storage room are not low enough and

the coils are not frosted to the ends, evidently the first thing to do is to find out why they cannot be made to carry frost throughout their entire length. The fact that a direct-expansion pipe accumulates frost indicates simply that the vapors and liquid ammonia passing through it are at a temperature sufficiently low to congeal the moisture of the air which comes in contact with it. So long as there is unevaporated liquid ammonia in contact with the vapor, the latter is said to be *saturated*, and the temperatures corresponding to the different back pressures can be readily determined by reference to the table XL of Properties of Saturated Ammonia.

If there is liquid ammonia enough at the expansion-valve, frost can be carried the full length of any coil and clear back to the machine, if desired, at a back pressure of 25 pounds, because the temperature of saturated gas at 25 pounds pressure is 11.5° F., which is 20.5° F. below the freezing-point of water. That a coil does not frost to the end under a back pressure of 25 pounds, indicates that either there is an insufficient supply of liquid ammonia at the expansion-valve, or that there is an obstruction which prevents a sufficient amount of it from passing the expansion-valve. An obstructed expansion-valve is indicated by there being little or no change in the sound of the passing liquid when the valve is opened several turns. Such obstructions can often be removed by the sudden opening and closing of the expansion-valve.

Scarcity of liquid at the expansion-valve can usually be recognized by an interrupted hissing sound, the hissing being caused by the passage of gas and the interruption by that of the liquid, maybe due to one of two things, viz., an insufficient charge of ammonia or too small a machine. If there is a sufficiently heavy charge of ammonia in the system and the machine is much too small, there will be no whistling sound heard at the expansion-valve, but the machine not being able to carry away the vapors of ammonia as fast as they are formed, the back pressure will rise higher until the extra pressure serves to retard the evaporation to such an extent that the machine cannot take care of it. It must also be remembered that as the back pressure rises the number of pounds of ammonia handled by the machine at a given pressure increases, because of the fact that the weight of a cubic foot of gas increases directly with the absolute pressure.

While the size of a machine cannot well be increased, its capacity for doing work may sometimes be increased by improving its efficiency. Sometimes low efficiency is due to dirt, which acts like an insulating material on the condensers and prevents the free radiation of heat; sometimes to insufficient or poorly distributed water on the condensers, and sometimes to so-called *permanent gases* within the condensation.

With well-sprinkled coils of ample size 210 pounds head-pressure is certainly too high for 59-degree water, and the trouble is liable to be due to any of the three causes above mentioned.

Ammonia as ammonia cannot deteriorate in quality, but at high temperatures, and, according to some authorities, more or less at moderate temperatures, it does slowly disassociate into its component gases, hydrogen and nitrogen. These gases, sometimes called *permanent gases*, because they do not liquefy, accumulate in the condenser, and, occupying the space that should be open to the ammonia, cut down the cooling-surface and thereby cause an abnormally high head-pressure. These gases should be purged from the system through a pipe or rubber hose, one end of which is connected to the purge-valve on the top of the condenser and the other immersed in a pail of water. If a sharp, cracking sound is heard and no bubbles rise to the surface of the water when the purge-valve is slowly opened it indicates that the gas is soluble in water and is ammonia. If, however, bubbles rise to the surface of the water the gas is proved to be comparatively insoluble and is not ammonia. The gases should be allowed to escape through the purge-valve into the water until no more insoluble gases appear. The water should be changed every few minutes, to keep it from becoming saturated with the ammonia, under which condition it will bubble through the water in much the same way as the permanent gases do, and may lead to deception regarding its true nature.

There should be enough liquid ammonia in the liquid-receiver at all times, so that no gas will pass the expansion-valve. The latter condition can be readily recognized by the temperature of the liquid-line between the receiver and the expansion-valve. It should be remembered that the temperature of the liquid ammonia going to the expansion-valve should be approximately that of the cooling water leaving the condensers, and that a wide variation in temperature

either way from that point would indicate an insufficient supply of liquid.

The condenser-coils should be kept clean and well covered with water at all times, and they should also be kept purged free from permanent gases.

CHARGING AND STARTING AN AMMONIA-COMPRESSOR

As each type of ammonia-compressor has its own individual features of construction, each particular machine will require special care and adjustment, so that no fixed rules can be laid down to suit all cases. There are, however, some general principles which are applicable to all types based on the compression system.

Before charging an empty machine with anhydrous ammonia all air must first be carefully expelled. This is done in various ways. One method often used is to pump the system full of gaseous ammonia and shut the engine down. Allow the water to flow in the condensers until all the ammonia in the system is condensed. The liquid ammonia, being heavier, will naturally gravitate to the bottom of the system. A valve can then be opened at the highest part of the system, and the pressure of the ammonia will force the air out; the presence of ammonia gas will indicate when to shut the valve. The system can then be allowed to stand another six or twelve hours, and the valve again opened. If there is any air remaining in the system, it will be driven out when the valve is again opened.

Before charging the system it can be thoroughly tested by working the compressor and permitting air to enter at the suction through the special valves provided for that purpose, and it should be perfectly tight at 200 or 250 pounds pressure per square inch, and should be able to hold that pressure without loss. While testing the system under air-pressure, it should be carefully and thoroughly cleaned of all dirt and moisture by blowing out.

In some cases it is impossible to eject all air from the plant by means of the compressor; therefore it is advisable to insert the requisite charge of ammonia gradually. Sometimes from 60 to 70 per cent. of the full charge is put in, and the air remaining in the system is allowed to escape through the purging-cocks with as little loss of gas as possible, subsequently inserting an additional quantity of ammonia

once or twice a day until all the air has been displaced and the complete charge has been introduced.

To charge the machine the drum of anhydrous ammonia is connected through a suitable pipe to the charging-valve. The machine should be run at a slow speed when sucking the ammonia from the tank, with the discharge- and suction-valves wide open. When one of the tanks is emptied the charging-valve is closed and another tank placed in position, and the process continued until the machine is sufficiently charged for work, when the charging-valve can be closed and the main expansion-valve opened and regulated. A glass gauge upon the liquid-receiver will show when the latter is partially filled, and the pressure-gauges, as well as the gradual cooling of the brine in the refrigerator and the expansion-pipe being covered with frost, will indicate when a sufficient amount to start working has been inserted.

The machine having been started and the regulating-valve opened, the temperature of the delivery-pipe should be carefully noted, and if it shows a tendency to heat, then the regulating-valve must be opened wider, while if it should become cold, the valve must be slightly closed, the regulation or adjustment thereof being continued until the temperature of the pipe is the same as the cooling water which leaves the condenser. If the charge of ammonia is insufficient, the delivery-pipe will become heated even when the regulating-valve is wide open.

Among the signs which denote the healthy working of the plant, beside the fact that it is satisfactorily performing its proper refrigerating duty, are the vibration of the pointers of the pressure- and vacuum-gauges (which clearly mark every stroke of the piston), the frost on the exterior of the ammonia-pipes (the liquid ammonia can be distinctly heard passing through the regulating-valve in a continuous stream), and the difference in temperatures between the condenser and the cooling water and the refrigerator and the brine.

CHAPTER XXI

THE ELEVATOR AND ITS WORKING

THE modern installation of elevator service has greatly increased the care and responsibility of the engineer, to whom such duties are usually assigned; and in view of these duties this chapter will be deemed not out of place, for not only the often complex details of the elevator but also those of the steam-pump or the electric motor are in charge of the engineer.

The direct-acting steam-motors for elevators are peculiar in their design, and require the care and watchfulness of the experienced engineer.

AIR-COMPRESSORS

The air-compressor—so much in use in operating mining-machines, hoists, conveyers, and air-locomotives, and for generating power for transmission for a variety of factory and operative purposes—becomes a specialty in the care of the engineer of such plants.

Of the many types or methods of operating elevators we note the following:

The direct-cable elevator, in which a reversible steam-engine winds and unwinds the rope-cable on a drum; the car, which is partly balanced by a cable and counterweight, with the stop- and reverse-valves operated by a lanyard, over which the car runs. The early safety-devices were a form of ratchet-stop (shown in Fig. 338), succeeded by friction-devices and speed-governors of many patterns.

Elevators of the type classed as hydraulic, and operated by water-pressure from a roof-tank or a pressure-tank fed by a steam-pump, are still in use. One of this type is illustrated in Fig. 339, and consists of a cylinder of one-half the length of the lift, with a piston and double piston-rods for safety. The pressure is downward on the piston, for elevating the car. A travelling sheave with the end of the car-cable fixed at the top gives the car twice the run of the piston. An auto-

matic stop controls the run of the car, and a lanyard-cable controls the speed by throttling the circulating-pipe.

A very compact hydraulic-elevator plant is detailed in Fig. 340, with a cylinder-capacity for a gear of 2 to 1 or 4 to 1, as desired. Here the pressure-tank is placed over the discharge-tank, with the steam-pump alongside. The principle of operation is contained in the action of the pilot- or transfer-valve, which is itself operated by a cable-lanyard passing through the car and over a wheel at the top of the shaft and over the valve-wheel seen at the top of the valve.

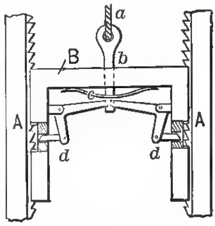


Fig. 338.—Elevator-stop.

When the car is at the top of the lift, the piston is at the bottom of the cylinder, with the valve closed to hold the car. To bring the car down, the valve opens the port of the transfer- or circulating-pipe, when the weight of the car and load transfers the water from above the piston to its under side, the velocity of transfer being regulated by the amount of opening of the valve. To start the car upward, the valve is moved past the stop-position and opens the exhaust-port between the bottom of the cylinder and the open tank, when the pressure from the high-pressure tank forces the piston down with a velocity regulated by the amount of opening in the exhaust-port.

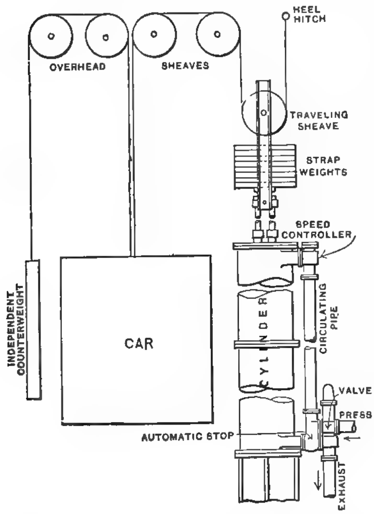


Fig. 339.—Hydraulic piston-elevator.

The duty of the pump is to transfer one cylinder full of water for each complete lift and return of the car, from the exhaust-tank to the high-pressure tank, in which the air is compressed to form the pressure-cushion.

Another type of the hydraulic system is the multiple effect of a pushing or pulling piston upon a series of pulleys, by which a short horizontal or vertical cylinder will produce a lift of many times the traverse of the piston, or plunger.

Fig. 341 illustrates a section and side view of the plunger type, in which A is the cylinder; P, the plunger; E_1, E_2, E_3, the three sheaves, which have their duplicate at the bottom and their anchor-eye for the cable at K, giving a lift of 8 to 1; H, the valve-chest, with the three positions of the lever for start, stop, and reverse. R is an automatic stop on a valve-rod operated by the arm Q on the sheave-frame.

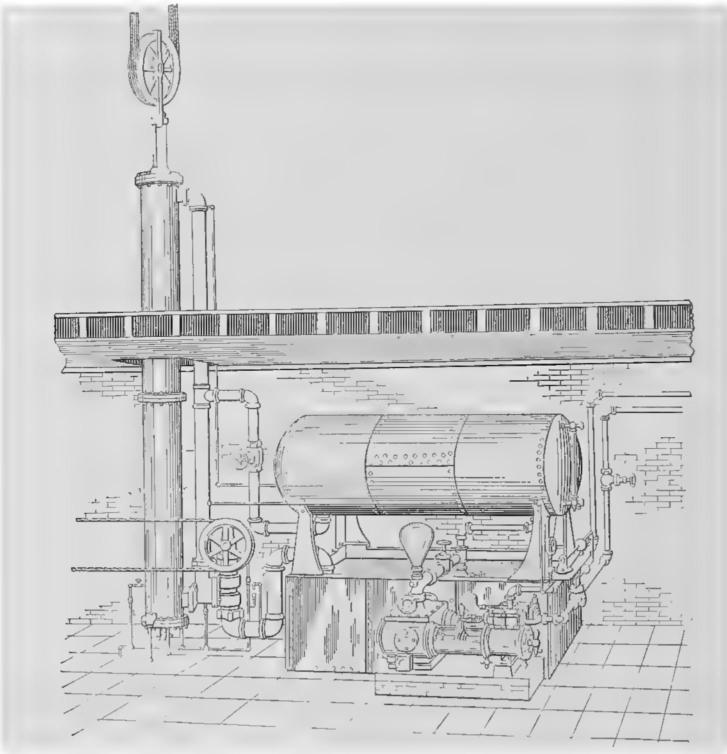


FIG. 340.—Pressure-tank elevator-plant.

The high-lift plunger-elevator (illustrated in Fig. 342) has been so perfected in its operation by experience with the failures of the telescopic lifts of the early hydraulic elevators that it has now attained a lift of 280 feet with a single plunger traversing a cylinder extending to a depth beneath the ground floor more than equal to the lift. These elevators run at speeds from 200 to 600 feet per minute; they carry a counterweight of 90 per cent. of the total load, and use a water-pressure of 185 pounds per square inch.

The elevators in the Trinity Building, New York City, are of this type, with plungers $6\frac{1}{2}$ inches diameter and with an upward friction of 500 pounds.

The total weight of the high-lift car, plunger, and fixtures is 8,460 pounds; full load, 1,600 pounds; total, 10,060 pounds. The counterbalance is 7,900 pounds, leaving 2,660 pounds, including friction, to be lifted by 6,000 pounds water-pressure under the piston—sufficient for a speed of from 400 to 600 feet per minute.

Valves entirely independent of the main controlling-valve are provided to bring the car to a gradual stop at each end of its travel. Two cables, one operating at the top of the run, the other at the bottom

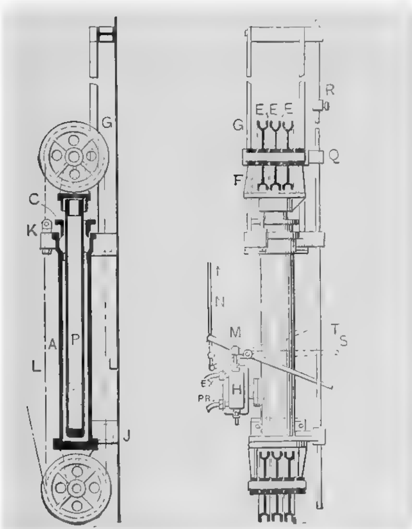


FIG. 341.—Plunger multiple lift.

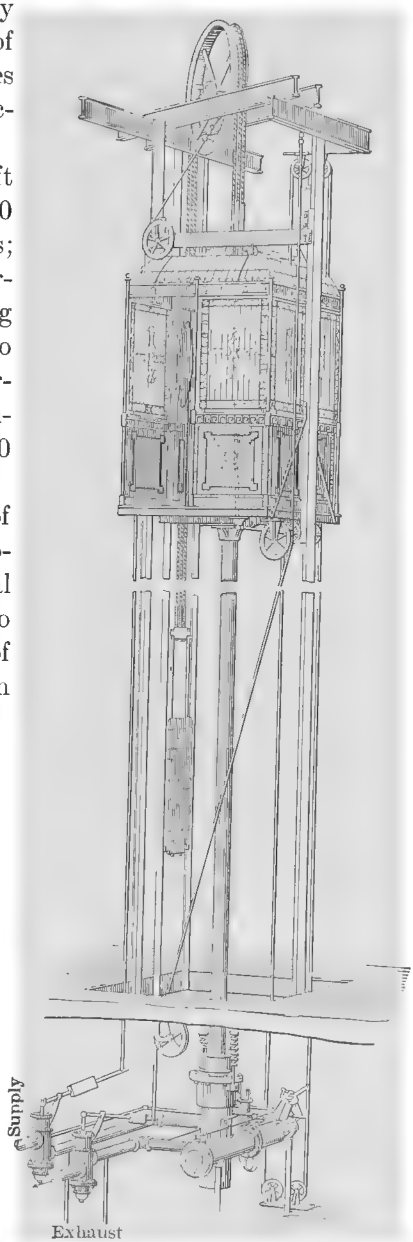


FIG. 342.—High-lift plunger-elevator.

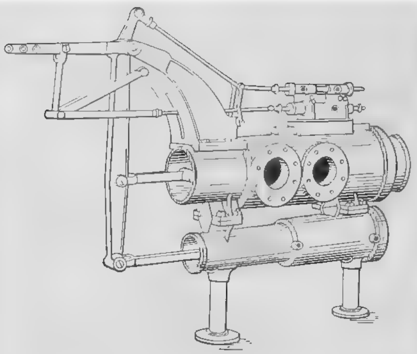


FIG. 343.—Three-way valve and pilot-valves.

of the run, are connected with these automatic valves, as shown in the cut (Fig. 342).

The overrun of the sheaves on the stop-cables causes their shortening, which lifts the weighted valve-levers and shuts off the supply-valve or exhaust-valve at the top or bottom of the car-run.

The elevator is controlled by a lever in the car, and the main three-way valve is operated by a pilot-valve, as illustrated in Fig. 343. In this way an easy and perfect control of the car is secured.

In Fig. 344 are illustrated the pushing plunger type and its operation as connected to the passenger-car, complete with governor and automatic stop, with lever and double sheave, on the valve-lanyard in the car for control.

This type is operated by the same combination of pump, open and closed tanks as shown in Fig. 340.

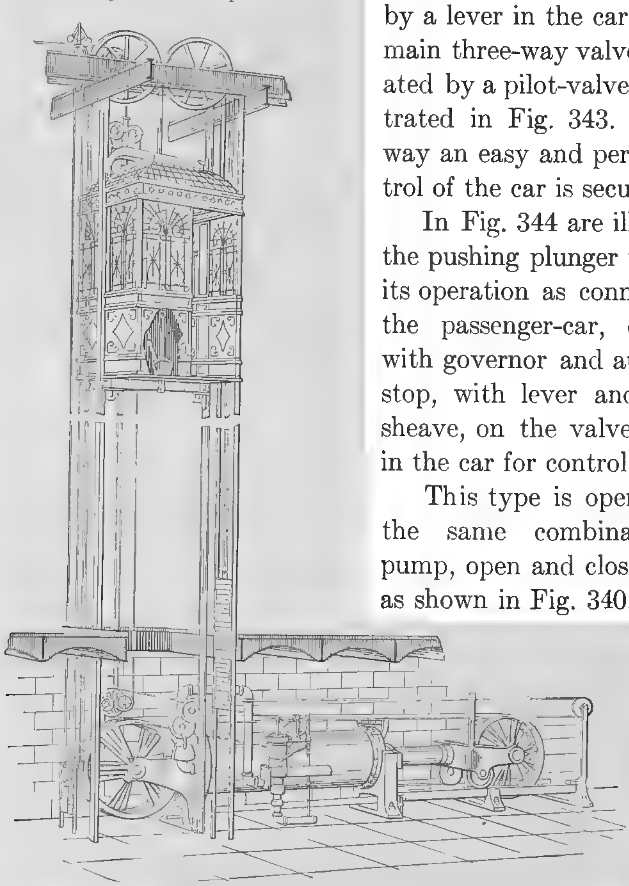


FIG. 344.—Hydraulic elevator, horizontal plunger.

In Fig. 345 is shown the safety-governor of the Otis elevator, by which a brake is applied to the governor-cable when the speed of the car exceeds the rate at which the governor is adjusted.

Its action is independent of the lifting-cables, so that in case of a breakage of the cables it will bring into action the car safety-devices to which it is connected, and will bring the car to a safe and easy stop. The governor-cable is endless, passing over the driving-sheave of the governor and a weighted sheave at the bottom of the shaft. It has a spring-stop connected to the gravity wedge mechanism under the car; it arrests its descent when excessive speed is attained from any cause.

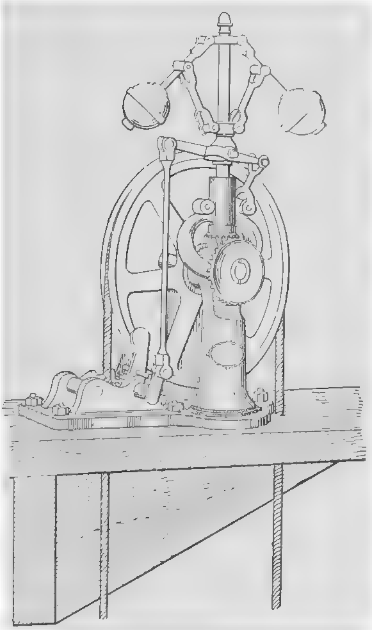


Fig. 345.—Otis automatic governor.

Fig. 346 shows the mechanism of the governor-cable connection to the gravity-wedge device. A and B show the cable running over the governor-sheave; to the side A are attached stops and a helical spring to ease the contact with its engaged lever on a sudden change of speed. The lever operates the arm of a rock-shaft that extends to the wedge-levers on each side of the car.

In Fig. 347 is represented the action of the gravity-wedge safety-device of the Otis Company. It is best described in their own words, which follow:

"Under the car is a heavy hard-wood safety-plank, on each end of which is an iron adjustable jaw, enclosing the guide on the guide-post. In this jaw is an iron wedge, withheld from contact with the guide in regular duty. Under the wedge is a rocker-arm, or equalizing-bar, with one of the lifting-cables attached independently at each extremity. The four lifting-cables, after being thus attached, pass over a wrought-iron girdle at the top of the car. Each cable carries an equal strain, and the breaking of any one cable puts the load on the other cables, which throws the rocker out of the horizontal position, and forces the

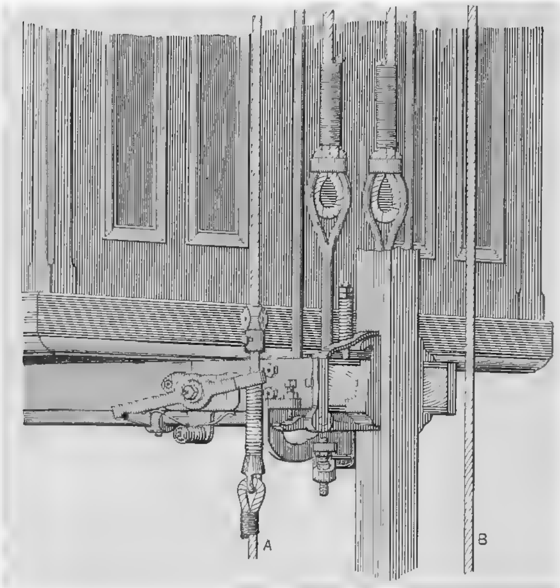


Fig. 346.—Automatic governor controlled safety-device.

wedges on both sides instantly and immovably between the iron jaws of the safety-plank and the side of the guides, stopping the car. It may be raised to any position by the unbroken cables, though it cannot be lowered until a new cable is put on."

A is the elevator car-guide; B, safety-wedge; C, safety-wedge shoe; D, adjustable gib; E, safety-wedge back spring; F, F, shackle-rods on ends of cables; G, equalizing-bar; H, lever to lift the wedge; I, I, set-screws on the equalizing-bar for adjusting the lever H to lift the wedge, by either movement of the equalizing-bar, from the breaking or stretching of any one of the cables.

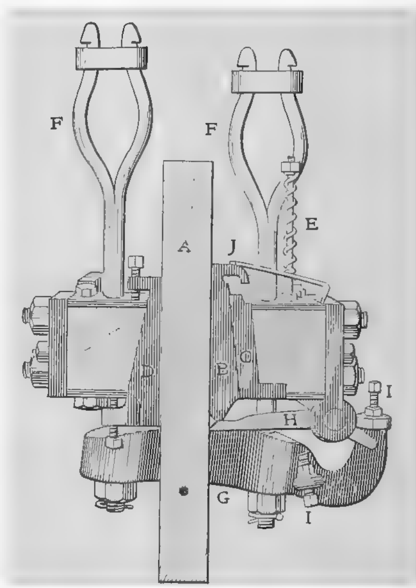
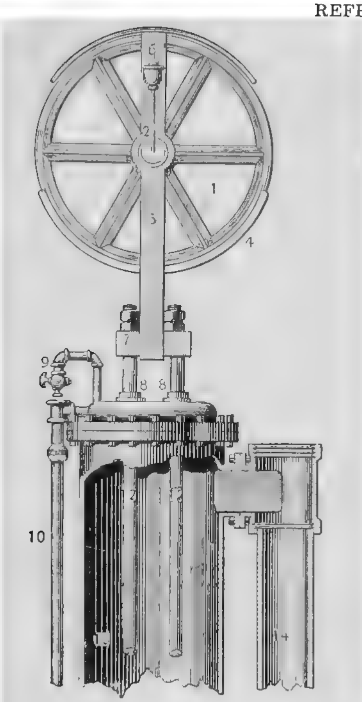


Fig. 347.—Gravity-wedge safety-device.

In Fig. 348 is shown a section of the Otis Company's vertical hydraulic cylinder, circulating-pipe, valves, and valve-gear. The operating-valve is balanced and moved by a rack-stem and a pinion on the shaft of the sheave carrying the car-lanyard. Pressure for operating is always full in the cylinder above the piston and in the circulating-pipe.

The valve, as shown in the cut,

REFERENCES TO NUMBERED PARTS



1. Travelling-sheave.
2 Travelling-sheave bushing.
3 Travelling-sheave pin.
4. Travelling-sheave guard.
5. Travelling-sheave strap.
6. Oil-cup.
7. Piston-rod cross-head.
8. Stuffing-boxes.
9. Air-cock.
10. Drip-pipe.
11. Curb on top head of cylinder.
12. Piston-rods.
13. Cylinder.
14. Circulating-pipe.
15. Piston.
16. Top follower.
17. Bottom follower.
18. Piston air-valve.
19. Piston-cup.
20. $\frac{3}{8}$-inch square rubber packing.
21. Set-screws for starting top follower when removing it to pack.
22 Cylinder-legs.
23. Drain from bottom of cylinder.
24. Water-chest.
25. Relief-valve, to relieve ram of water when the valve is suddenly closed during the ascent of car.
26. Valve-chamber.
27. Valve-plunger, consisting of: A. Rack-follower; B. Valve-stem; C. Top to valve piston-cup; D. Bottom to valve piston-cup; E. Spider; F. Valve-cup packings.
28. Valve-rack.
29. Valve-rack shoe.
30. Valve-pinion shaft.
31. Valve-cap.
32. Valve-glands on pinion-shaft.
33. Valve-sheave.
34. Check-valve.

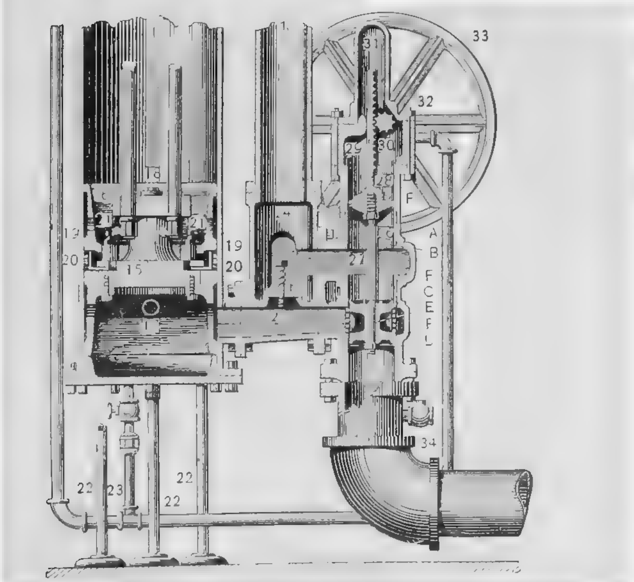


FIG. 348.—Section of hydraulic-elevator cylinder and valves.

is at " stop " for the car; lowering it opens communication between the upper and lower sections of the cylinder, and the car descends by its own weight and by the transfer of the water from above to below the piston. By raising the valve the water beneath the piston discharges, and the higher pressure on the upper side of the piston sends the car upward.

One of the later innovations in the elevator line has been brought out in the ramp, or escalator, a contrivance which affords a convenient way of getting upstairs. One of the earlier devices is shown in

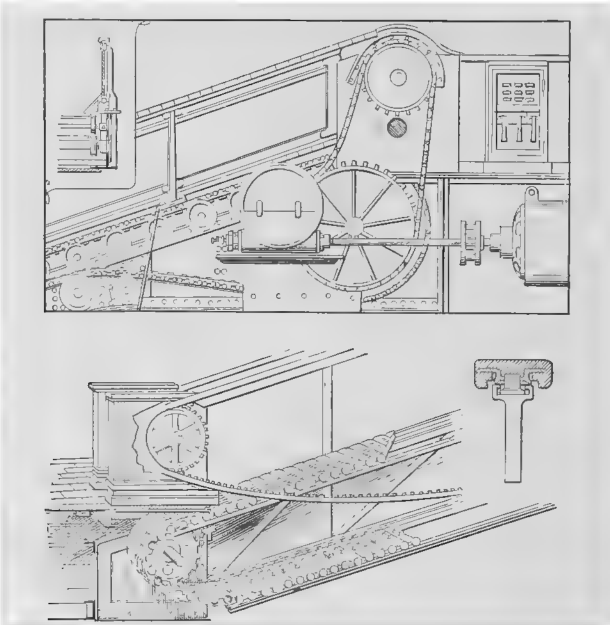


Fig. 349.—The ramp.

Fig. 349, with sections of the upper and lower ends of the ramp with the driving-gear. A dynamo and a transmission device drive the upper drum and guards at a mean speed of 20 inches per second.

The system comprises an endless web formed of bars of wood which are provided with rollers that are formed of a material called " hemacite " and that run upon rails. The returning half is suspended from a rail lodged in the lower chord of the principal girder. This arrangement of chains with detachable links permits of doing away with stretchers.

The jointed web is actuated by a chain of which each link corresponds to one of the bars of wood. This passes at the upper part over an indented wheel actuated by the electric motor, with the interposition of a shaft with a ratchet to prevent any return in an opposite direction.

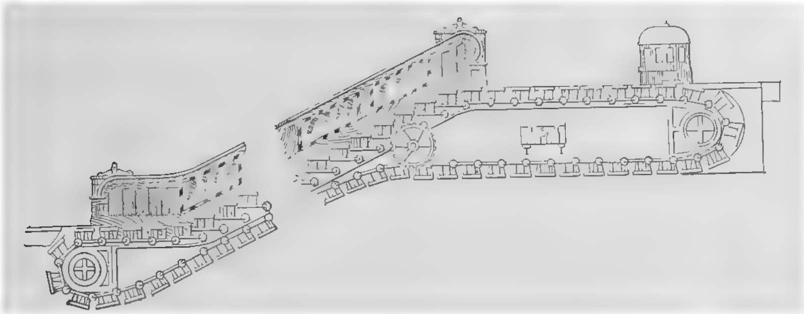


FIG. 350.—The step-escalator.

The jointed bars are provided with rubber projections for the purpose of giving the feet a firm hold. These projections, which are arranged in longitudinal bands, make their exit at the lower part and disappear at the upper between the teeth of metallic combs designed to take up and set down the passengers without jerks. The guards consist also of endless chains covered with rubber and cloth. Each link of the chain slides in a groove that prevents any lateral displacement.

A perspective view of the lower end of the ramp in the lower section of the cut shows the jointed web, sprocket-drums, and hand-rail.

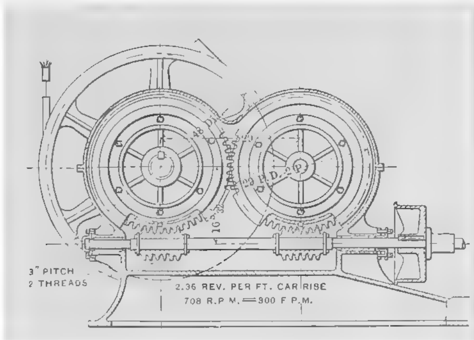


FIG. 351.—Worm-gear elevator.

In Fig. 350 is illustrated the newest type of escalator, brought out by the Otis Company, and in use on the Sixth Avenue Elevated Railroad and at the Macy department store in New York City. It will be seen that in this type the passengers step onto the escalator on an even moving floor that rises into steps at the incline, which again form an even floor at the top for a sufficient

distance to step off without trouble or danger. The hand-rail travels at the same rate as the steps. The capacity ranges from 4,000 to 6,000 persons per hour.

Many direct-cable elevators are driven through worm-gear which has its own drawbacks from wear and cutting of the gear. For safety in this respect the double worm-gear is in use, which reduces the friction, serves the purpose of balancing the thrust of the driving-shaft, and is also a means of safety from breakage of teeth. The worms have right-and-left-hand threads. The Sprague type of electric-driven elevator is illustrated in Fig. 351.

THE MASON ELEVATOR PUMP-PRESSURE REGULATOR

This regulator, which is illustrated in Fig. 352, is designed for the larger sizes of steam-pumps operating hydraulic elevators. The important feature in this machine is that it operates on the slightest change of pressure, opening and closing the steam-valve to its fullest extent promptly and with certainty.

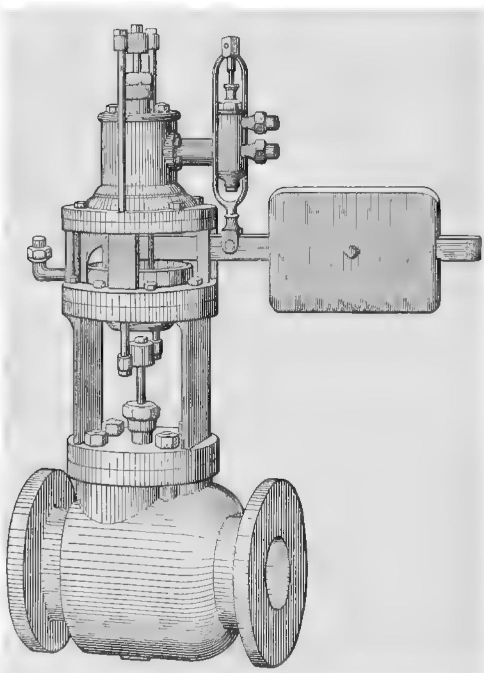


FIG. 352.—Pump-pressure regulator.

Referring to the sectional view, Fig. 353, the operation is as follows: Steam from the boiler enters the regulator at the inlet, and passes through the main valve into the pump, which continues in motion until the required water-pressure is obtained in the elevator system, which acts, through a $\frac{1}{4}$-inch pipe connected at A, upon the diaphragm B. This diaphragm is raised by the excess water-pressure, and carries with it the weighted lever, opening the

auxiliary valve D, and admitting the water-pressure from the connection E to the top of the piston, at the same time opening the exhaust-ports under the piston, thus allowing the water under the piston to escape into the drip-pipe, thereby pushing the piston down, which closes the steam-valve and stops the pump.

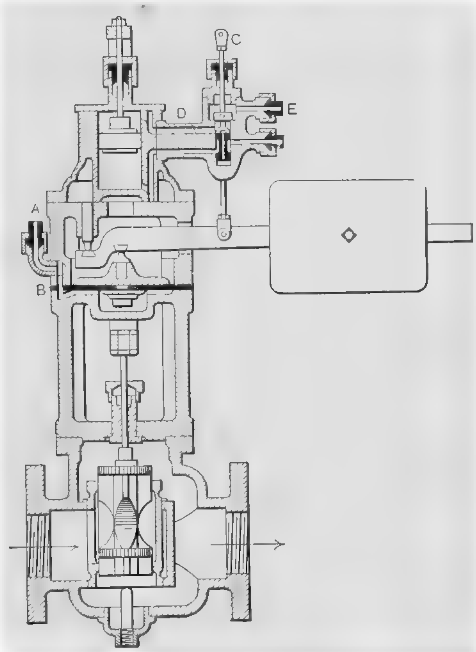


FIG. 353.—Section of pump-pressure regulator.

As soon as the pressure in the system is slightly reduced, the lever, on account of the reduced pressure under the diaphragm, is forced down by the weight, carrying with it the auxiliary valve, thus opening the exhaust to the top of the piston, and at the same time admitting the water-pressure under the piston, which is now forced up and opens the steam-valve, again starting the pump.

The main balanced valve and the controlling-valve are connected by an outside yoke, as are also the auxiliary valve D and the lever, as shown in Fig. 352.

AIR-COMPRESSORS AND COMPRESSED AIR

The steam end of an air-compressor is essentially the same, in all its details, as that of other steam-engines, as explained in previous chapters of this work. The air end, and its action and operation, come within the province of the engineer, and require some consideration. In many places the distribution and use of compressed air also require some knowledge on the part of the engineer of its properties and action. For details of the uses and work of compressed air for all purposes, the author recommends reference to his large work on

"Compressed Air," published by the N. W. Henley Publishing Company, New York City.

Compressed air is not only used for running local motors, hoists, and rock-drills, but is largely in use for refrigeration in the marine and

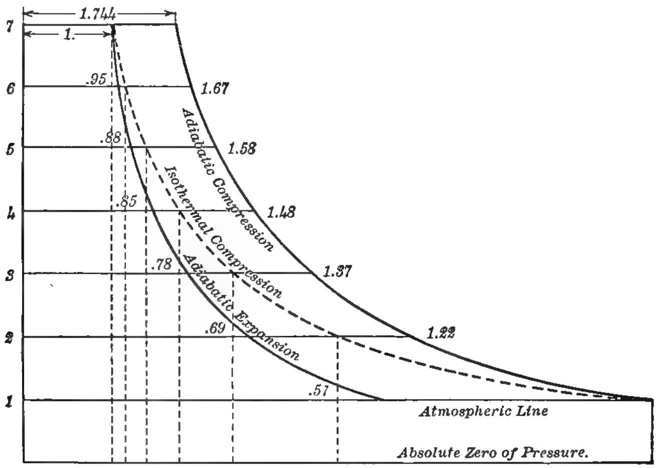


FIG. 354.—Diagram of compression and expansion of air.

the naval service. The compressed-air brake is at the fore in railway service.

By compression and expansion air obeys the laws of thermodynamics, becoming hot by compression and cooling by expansion. For an assumed compression and expansion without change of temperature—isothermal—its volume and pressure are in exact inverse proportion, but in actual practice in the compressor and motor the lines of pressure and expansion, as shown on an indicator-diagram, are adiabatic to meet the conditions of temperature.

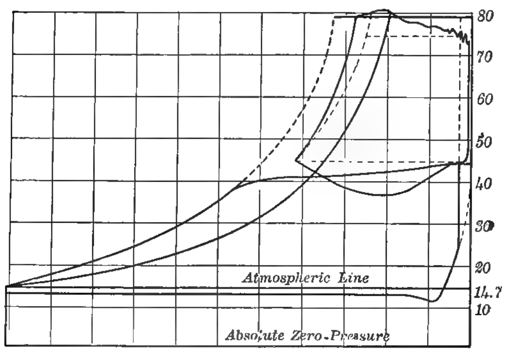


FIG. 355.—Two-stage compression.

In the diagram (Fig. 354) are shown the theoretical curves due to compression and expansion where there is no transfer of heat to or from the walls of the cylinder. The figures along the margin of the curves show the change of volume from the isothermal line. In actual practice the compression-volumes are less, and the expansion-volumes are somewhat greater, than shown by the figures.

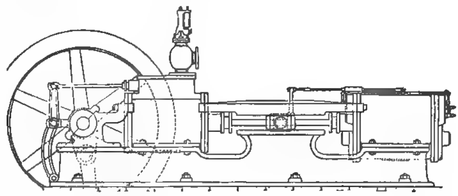


FIG. 356.—Bennett air-compressor.

The mean pressure due to compression and expansion, as taken by an indicator, can be figured in the same manner as with the steam-card (see Indicator, Chapter XIII), and needs no further explanation. Full details of the theory, practice, and work of compressed air are given in the work on "Compressed Air and Its Uses" by the author.

The effect of compressing air in compound or by two stages is shown in Fig. 355, and for high pressures a three-stage compression shows much economy in the power used for compression.

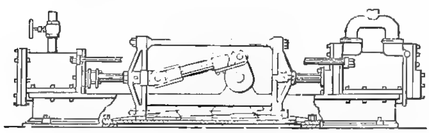


FIG. 357.—Clayton compressor.

In the two-stage diagram it may be seen that the lower curve is that of the isothermal up to 80 pounds per square inch, while the upper curve shows the increased volume due to compounding with an intercooler to shrink the volume before it enters the second cylinder. In this way the economy in power by compound compression up to 100 pounds is about 15 per cent., and increases with higher pressures.

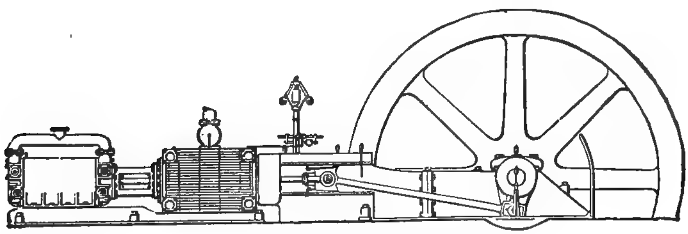


FIG. 358.—Corliss air-compressor.

The accompanying illustrations are those of some of the models and details of compressors in use.

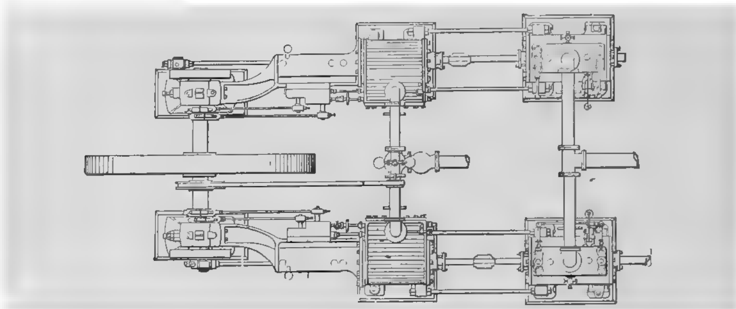


Fig. 359.—Duplex tandem air-compressor.

In Fig. 356 is shown an elevation of the Bennett air-compressor, with direct piston-connection, cross-head, and outside connecting-rods to the crank-pins in the fly-wheels. The eccentric on the shaft at the rear of the steam-cylinder is linked to a vertical lever and valve-rod.

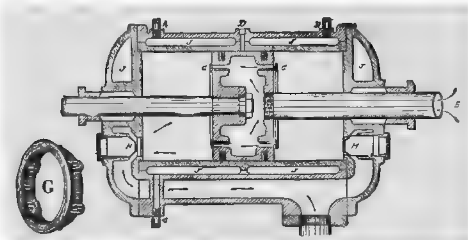


Fig. 360.—Ingersoll-Sergeant cylinder.

In Fig. 357 is shown the elevation of an air-compressor of the Clayton type, in which the cylinders are placed at each end of the bed frame, and with yoked piston-rod connection and with the crank and connecting-rod within the yoke.

The direct-connected tandem system, with a Corliss steam-cylinder and centrifugal governor, is shown in Fig. 358. It is a type of air-compressor now rapidly increasing in economy and usefulness by tandem compounding and cross-compounding, and is in use in large plants.

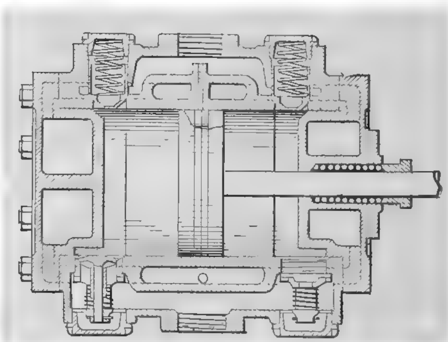


Fig. 361.—Vertical lift-valve cylinder.

An example of the duplex tandem type of air-compressor is shown in the plan (Fig. 359). In this type the steam

end is provided with a throttling-governor and riding-cutoff for each cylinder.

The air-cylinders are of the Ingersoll-Sergeant pattern, set on a sole-plate and fastened by rods to the steam-cylinders. The piston-rods are connected by couplings, and the air-supply is regulated by a governor.

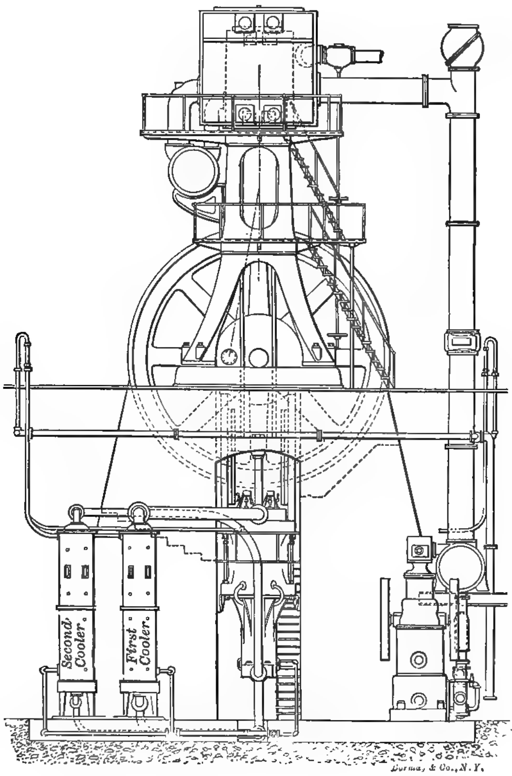


FIG. 362.—Vertical four-stage air-compressor.

Fig. 360 shows the cylinder, piston, and valves of the Ingersoll-Sergeant pattern. It has a through hollow piston-rod, into which the air is drawn to feed the hollow piston and cylinder through the annular valves, one of which is shown at G. These valves open and close by their momentum, and are free from obstructive pressure against the incoming air.

Fig. 361 illustrates a section of an air-cylinder, with vertical lift-

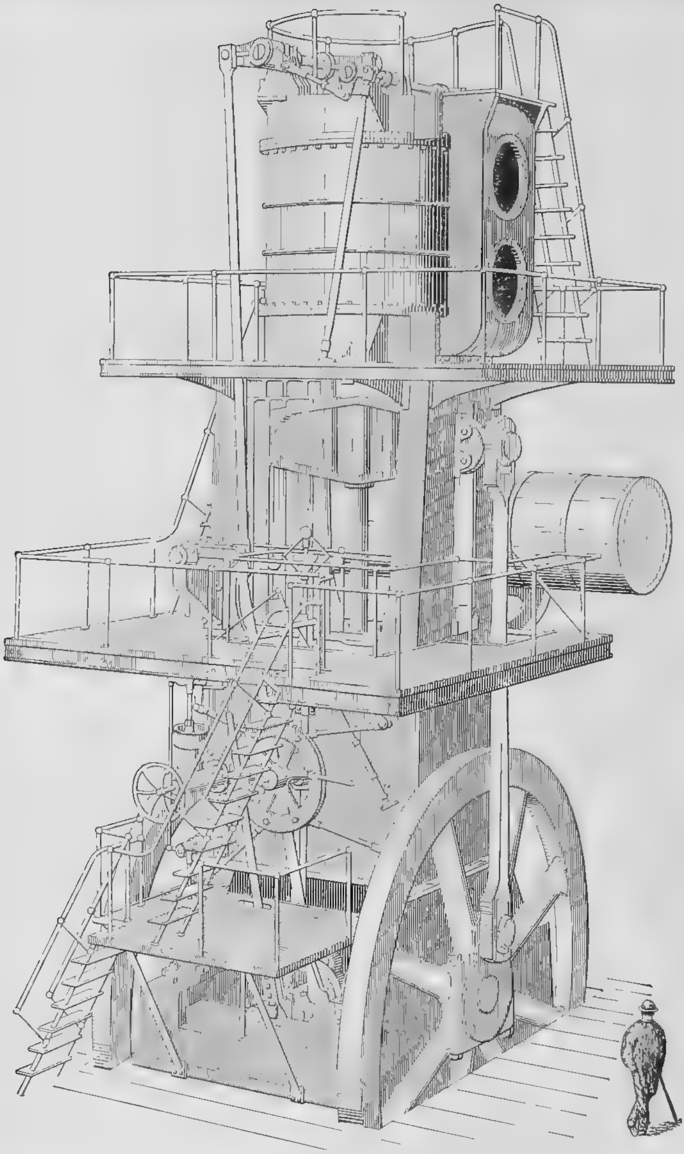


FIG. 363.—Reynolds-Corliss blowing-engine.

Steeple type, condensing; long cross-head connections to piston-rods and crank-rods. The air-cylinder has mechanically operated valves. Built by the Allis-Chalmers Company for blast-furnaces, smelters, and Bessemer work.

valves controlled by springs, a solid piston, and with cylinder-heads water-jacketed.

In Fig. 362 is shown an end-view sketch of the largest high-pressure air-compressor ever built. The steam-power of the compressor is derived from a duplex vertical cross-compound engine with Reynolds-Corliss valve-gear. With steam-pressure of 150 pounds and 40 revolutions per minute, it is equal to 1,000 horse-power. Directly beneath each pair of steam-cylinders is placed a pair of air-cylinders, tandem, and connected to the steam-cylinder cross-heads by a yoke-frame. The steam-cylinders are 32- and 68-inch by 60-inch stroke. The air-cylinders are 46-, 24-, 14-, and 6-inch by 60-inch stroke; they are tandem in pairs and single-acting. The approximate capacity at the above speed is 2,269 cubic feet of free air per minute. The pressure in the first cooler is 40 pounds; second cooler, 180 pounds; third cooler, 850 pounds, and in the after-cooler 2,300 pounds. It was built by the Allis-Chalmers and Ingersoll-Sergeant companies for the Metropolitan Railway Company, New York City, for charging the car-tanks and operating air-power cars.

There is much acumen required in an engineer operating a large air-plant that is not usual in the experience of the young engineer, so that a special study should be made of the written or personal instructions given by the builders of such plants, as their construction is as variable in detail as that of steam-plants.

A type of the massive engines used for supplying air under pressure to the blast-furnaces of the iron industry is shown in Fig. 363. The man on the floor represents a comparative proportion for the size of this colossal blowing-engine.

CHAPTER XXII

THE COST OF POWER-ECONOMY

THE subject of the cost of power for various mechanical uses and for electric power and lighting has been a theme of engineering papers and of discussion in technical journals for many years past, with varying results depending upon the varying conditions in the cost of material and labor.

We append an abstract from a communication of Mr. William O. Webber, of Boston, Mass., containing his experience in the matter of the cost of a steam-plant and the operating cost of plants of various sizes and types. The cost of land and buildings will probably make a material difference in estimating the total cost of power, and for the annual cost of operating, the insurance, interest, and repairs should enter into the items of expense.

The following table shows the estimated cost of a plant in the Eastern States for a 60-brake horse-power:

TABLE XLI.—COST OF A 60-BRAKE HORSE-POWER PLANT

Land for engine and boiler-room		$2,500.00
Buildings for engine and boiler-room		2,500.00
Chimney		1,200.00
80 horse-power boiler	$790.00	
Ash-pan for boiler (below high tide-level)	120.00	
Blow-off of sink	31.00	
Boiler- and engine-settings	1,282.00	
Damper-regulator	75.00	
Injector-tank	10.00	
Water-meter	60.00	
Piping	22.13	
Pump	146.50	
Feed-water heater	70.40	
Pipe-covering	70.75	
		2,677.78
Engine 12 x 30	1,065.00	
Pan for engine-flywheel	72.00	
Steam-separator	60.00	
Oil-separator	41.80	
Piping, freight, and cartage	1,026.41	
		2,265.21
Shafting in place	550.00	
Belting in place	285.00	
		835.00
		$11,977.99

$\frac{11,977.99}{60}$ = $199.61, or say $111 per brake horse-power for the machinery alone.

TABLE XLII.—COST OF STEAM HORSE-POWER PER 1 BRAKE HORSE-POWER PER ANNUM.

William O. Webber.

	SIMPLE HIGH SPEED.				COMPOUND CONDENSING.										TRIPLE EXPANSION.	
Size of plant, horse-power	20	40	60	80	100	200	300	400	500	600	700	800	900	1,000	1,500	2,000
Cost of plant per horse-power, dollars	200.00	190.00	180.00	175.00	170.00	146.00	126.00	110.00	96.00	85.00	76.00	69.00	64.00	60.00	58.00	56.00
Fixed charges, at 14 per cent	28.00	26.60	25.20	24.50	23.80	20.40	17.65	15.40	13.45	11.90	10.65	9.65	8.95	8.40	8.12	7.85
Coal per horse-power in pounds	12.00	10.00	9.00	8.00	7.00	6.50	6.00	5.50	5.00	4.50	4.00	3.50	3.00	2.50	2.00	1.50
Cost, at $4.00 per ton	66.00	55.00	49.50	44.00	38.50	35.70	33.00	32.00	27.50	24.70	22.00	19.20	16.50	13.75	11.00	8.25
Attendance, 10-hour basis	30.00	20.00	15.00	13.00	12.00	10.00	8.60	7.25	6.20	5.40	4.70	4.15	3.75	3.50	3.25	3.00
Oil, waste, and supplies	6.00	4.00	3.00	2.60	2.40	2.00	1.72	1.45	1.24	1.08	.94	.83	.75	.70	.65	.60
With coal at $5.00 per ton	146.50	119.35	105.07	95.10	86.40	77.10	69.22	61.90	55.29	49.28	43.79	39.73	34.05	29.80	25.77	21.75
" " " $4.50 " "	138.25	112.47	98.80	89.60	81.50	72.60	65.07	58.10	51.79	46.18	41.04	36.28	32.00	28.05	24.39	20.72
" " $4.00	130.00	105.60	92.70	84.10	76.70	68.10	60.97	56.10	48.39	43.08	38.09	33.83	29.95	26.35	23.02	19.70
" " $3.50 "	121.75	98.72	86.51	78.60	71.90	63.70	56.82	50.50	45.04	39.98	35.54	31.48	27.87	24.60	21.64	18.67
" $3.00	113.50	91.85	80.32	73.10	67.00	59.20	51.67	46.70	41.49	36.88	32.79	29.03	25.80	22.90	20.27	17.65
" $2.50 "	105.25	84.97	74.13	67.60	62.30	54.75	48.59	43.00	38.09	33.83	30.04	27.18	23.75	21.20	18.89	16.60
" $2.00 "	97.00	78.10	67.95	62.10	57.45	50.25	44.47	40.10	34.64	30.73	27.29	24.23	21.70	19.47	17.52	15.57

The above and following tables are the result of late experience, and include the average cost of land and buildings necessary for the steam-plants upon which the reports of steam-power plants for factories have been based. The approximate ratio is 1 indicated horse-power = .85-brake horse-power for small powers; less for the large power-plants.

The annual cost per brake horse-power, of operating the ordinary types of steam-plants under 200 horse-power may be stated as follows:

TABLE XLIII.—OPERATING EXPENSES.

Average horse-power.	Cost of coal per ton, 2,240 pounds.	Total cost per annum.	Remarks.
182	$3.50	$57.59	No land or building costs.
133	3.25	60.00	" " cost.
100	3.50	65.00	" " or building costs.
97	4.45	86.80	All costs included.
75	2.90	92.40	No land or building costs.
50	4.75	111.05	" " " " "
20	4.45	133.50	All costs included.

EFFECT OF LOAD-FACTOR ON THE COST OF POWER

For an electrical system the great desideratum is a high load-factor with consequent greatest return on investment, load-factor being the ratio of average to maximum load. All the factors of expense included in cost of power to the consumer are then operating at maximum economy, and cost of power is at a minimum.

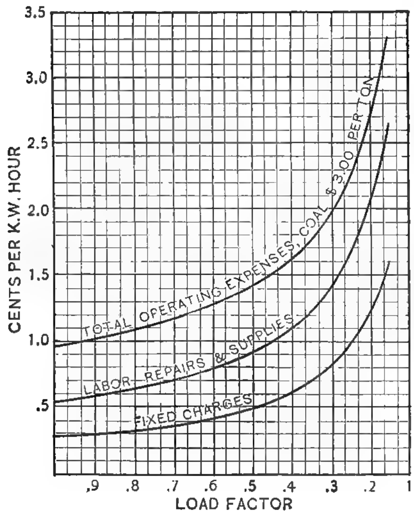


FIG. 364.—Operating expenses of a 900-kilowatt condensing steam-plant.

In Fig. 364 is a diagram showing the total operating expenses, labor repairs and supplies, and fixed charges in curves representing the cost in cents per kilowatt hour for full-load and under-load conditions.

In Fig. 365 is also a diagram for the same conditions in a non-condensing plant, in which it may be observed that the two lower curves are the same as those in Fig. 364, and that the uppermost curve represents the increased cost due to the steam end of the plant.

Lighting of residences and offices produces a peak in the late after-

noon and evening, with but little load the remainder of the twenty-four hours; consequently the average load on the plant with lighting only is small and the load-factor low. A commercial motor-load in connection with lighting will increase the average load even though causing a greater peak. The addition of a street-railway load still further increases the day load, but in consequence of the heavy-demand load during the rush-hours, when the public is going to and from business, which occurs at the peak of the lighting-load, the peak-load on the plant is greatly increased. This heavy peak, with but a small average load over the twenty-four hours, produces a low load-factor, as a portion of the machinery is shut down the greater part of the time.

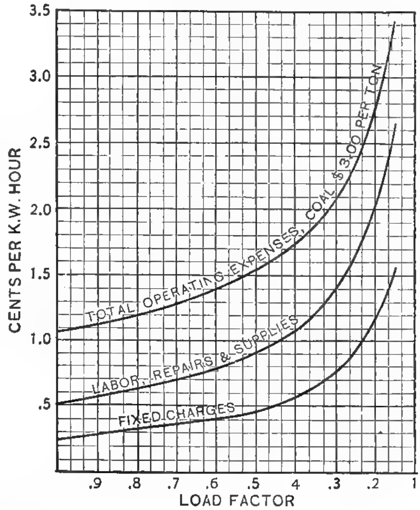


FIG. 365.—Operating expenses of a 900-kilowatt non-condensing steam-plant.

In the cost of power to the consumer various expenses are involved, viz., management, distribution, and production.

For a given system with given peak-load the cost of management is practically constant, no matter what the load-factor.

Cost of distribution is constant with various load-factors in so far as the fixed charges and maintenance are concerned. The losses in distribution, however, vary, these consisting of losses in lines, transformers if alternating current is used, meters, losses in grounds, and losses from theft of current.

As to cost of production, the higher the load-factor the greater is the amount of power produced, the longer does the apparatus operate at best efficiency, the lower the ratio of fixed charges to total operating expenses, and consequently the lower the cost of power per unit.

ECONOMICAL SUGGESTIONS IN THE GENERATION AND USE OF STEAM FOR POWER PURPOSES

Although it is difficult to give any general information on this subject which will be of use or interest in the great variety of particular cases, it may be of some interest, and possibly of assistance to those who are managing or operating power-plants, to discuss some of the principles upon which economy in the use of steam depends.

Beginning with the boiler, which is the first step in the production of power from fuel, it may be laid down as a good rule that it is more economical to use boilers of reasonably large size than to subdivide into a larger number of small units. The length and area of grate that can be conveniently fired or kept evenly covered with coal are, perhaps, the limiting features, if hand-firing is to be used. Working from this rule, a grate should not be over 7 feet long or more than 5 feet wide, which would give 35 square feet of grate-surface. The quantity of coal that may be burned on such a grate varies widely with the kind of fuel and strength of draught. Using bituminous slack coal of fair quality, with good natural draught or moderate-induced draught, it should be possible to burn 25 pounds of coal per square foot of grate per hour, or 875 pounds of coal per hour for 35 square feet, and if this coal will evaporate say 8 pounds of water per pound of coal, the boiler, if constructed with heating-surface in proper proportion, would evaporate 7,000 pounds of water per hour, which would be equal to a little over 200 standard boiler horse-power. In order to give good economy, the boiler should have from 2,000 to 2,400 square feet of heating-surface to evaporate this quantity of water economically. The return tubular boiler, on account of the amount of tube-surface in proportion to the direct surface exposed to the fire, should have not less than 12 square feet per horse-power; the water-tube type, from 10 to 11, and the internally fired type, which has a larger amount of direct heating-surface in the furnace and tubes than either of the others, should have 9 to 10. If the grate-surface is larger than that described, probably the grate will not be evenly covered with coal, or the fire will be dead in spots, so that too much cold air will pass through.

The economy in burning fuel is a matter requiring great skill and experience, and depends entirely upon the evenness, thickness, and condition of the fire, which controls entirely the air-supply and, therefore, the perfection or imperfection of the combustion.

It is too often the case that the demands for increased horse-power are met by grate-surface too large in proportion to the heating-surface of the boiler or forced draught, and too little attention is given to careful firing, with heating- and grate-surfaces in proper proportion to give best economy, and frequently a great deal of money is spent in obtaining high-class engines and condensers, whereas the principal loss is in the boiler and fire-room.

The question is often asked whether in case of installing a certain horse-power of boilers, say 300 horse-power, it would be more economical to have three boilers of 100 horse-power each or two boilers of 150 horse-power each. We would advise to have the two larger units, as it will always be found that the larger boilers have less radiation, less air-leakage, and better combustion than a corresponding horse-power in small units. If it is necessary to have a spare unit for cleaning, let there be another one provided of the same size.

In regard to the pressure to be carried, it is well known that a high pressure gives a greater amount of expansion and better economy in proportion to the fuel burned. Even with simple engines in which it is not possible to obtain the full advantage of expansion, the high pressure of steam, which is drier and contains a larger number of heat-units in proportion to the volume, gives the best results. Every boiler should be designed for not less than 150 pounds pressure per square inch. Even if it is not possible to utilize the full pressure, the boiler will be stronger, last longer, and be a better investment in the long run. In this respect the water-tube, boiler, or some form of internally fired boiler in which the shell-plates are not exposed to the high temperature of the furnace, is certainly safer than the horizontal return tubular boiler, because for large units intended to carry high pressure the shell-plates and seams must be of considerable thickness, which, being directly exposed to the hottest part of the fire, are liable to give trouble, especially if there be any scale or sediment in the water which may settle on the bottom directly over the fire.

As to the economy of various types of boilers, experience shows that all of the standard types—horizontal return tubular, water-tube,

or internally fired—if they are designed with proper proportions of heating- and grate-surface, give about the same evaporation per pound of coal, provided they are in good condition and clean both on the fire- and on the water-surface. While the externally fired boilers, either of the return tubular or the water-tube type, are said to have some advantage in combustion, on account of the heat of the brick furnace, they are subject to losses which are more serious in the way of air-leakage in the setting and radiation.

The repairs and cost of keeping up brick furnaces are considerable, and as a result of deterioration there is more or less air-leakage through the brickwork going on constantly. In this respect the internally fired boiler has a great advantage over return tubular or water-tube boilers with brick furnaces, as it will be just as efficient after continued use as when first started.

In any type of boiler it is of great importance to keep the tubes and other surfaces free of soot and scale. Otherwise a large loss may be sustained. It is a mistake to depend entirely on the steam-blower or tube-cleaner, which only removes the loose soot, a scraper being necessary for occasional use to free the hard scale; otherwise it will in time accumulate on the fire-surfaces. It is necessary to point out that scale, or, worse still, oil on the inside of a boiler may be a source of great loss, experience having proved that even a thin film of oil will so prevent the transfer of heat that the plates or tubes will be burned in a short time. Nothing but pure water should be used for making steam, and the practice of making the boiler do duty as a water-purifier as well as a steam-generator cannot be too strongly condemned. If the owners of steam-plants could realize that a very small deposit of soot on the outside and scale on the inside mean a loss of from 10 to 20 per cent. of the total fuel-consumption, they would be convinced that it would be much cheaper to spend money in purifying apparatus, so that the scale or sediment will be removed before the water is fed to the boiler.

The next step to be considered is the heating of the feed-water. This may be accomplished in two or three ways: first, by means of the exhaust-steam, which, coming from a non-condensing engine, is capable of heating the feed-water to 212° F. and of saving say 12 to 15 per cent. as compared with feeding cold water. For large plants where it would be advisable to use induced draught to make up for the

loss in temperature of the chimney-gases, which produce the draught, it will undoubtedly pay to use an economizer; but as this apparatus is expensive both in first cost and in up-keep, the amount saved in utilizing the waste gases from a small plant would probably not offset the outlay. The closed type of feed-water heater is about as efficient as the open type, provided the water is pure, and it avoids trouble from pumping hot water, but the open type is frequently made use of to assist in purifying the water and, if properly managed, may give good service in that respect. For condensing-engines a primary heater of the closed type may be installed between the engine and condenser, which will help to condense the steam and heat the feed-water to a low temperature—say 130 to 140° F. A secondary heater, either of the closed or the open type, may be used to heat the feed-water to a still higher temperature, say 212° F., by the use of the exhaust from the feed- and air-pumps, which exhaust cannot be used more profitably than in this way, as all the heat is returned to the boiler.

In regard to the type of engine used for the plants, if the size of the plant is sufficient, and the work comparatively steady, the highest possible results may be obtained from compound condensing-engines using the highest possible pressure of steam, but under other conditions, such as variable load or low pressure of steam, it may be quite possible that the simple engine will give better results and cost less for repairs. With low steam-pressure, non-condensing, there is certainly nothing better or more economical than a single-cylinder Corliss engine where it can be installed to advantage. In the case of direct-driven electric units of small size, it is necessary to use high- or medium-speed engines, both on account of the loss in friction that would result if counter-shaft and belting are used and because the higher-speed machines will give the best regulation. For small units up to say 75 or even 100 horse-power, there is nothing better than the modern high-speed automatic engine, provided it is of good design, not overloaded, and not overspeeded.

As illustrating the slight wear of high-speed engines under favorable conditions, a Robb-Armstrong engine of 12-inch stroke, which has been running at 275 revolutions per minute for electric lighting for twelve or fourteen years, shows only about two-thousandths of an inch wear of the journals and six-thousandths of an inch wear of the shaft-bearings.

Unfortunately, this class of engine is so frequently overloaded and overspeeded that it gives poor results and gets a bad name, whereas the Corliss slow-speed type of engine is limited both in the matter of speed and horse-power, because the cut-off of the single-eccentric type will not go much beyond half-stroke, and in that way the engine is saved from overloading and abuse, and this is, perhaps, one of its many advantages. A compound engine is not suited to low pressure or irregular loads, and the extra cylinder and complication of parts are a great objection under such conditions. When a condenser is used, even with low pressure and somewhat irregular loads, it may be employed to advantage, whereas with high pressure, say from 125 to 150 pounds or over, the non-condensing compound will give the best results, unless the load is very irregular and running to light loads a large part of the time.

The question is sometimes asked whether it pays to reduce the pressure when the load is light. From experience, we do not believe it pays to reduce the pressure on the boiler, excepting in very extreme cases, but if it can be done by throttling before the steam reaches the cylinder of the engine, it would be an advantage, because this retains the heat-units due to the higher pressure in the steam and the throttling has a slight superheating effect.

Another source of considerable loss in the operation of steam-plants, particularly large ones, are the insufficient size of piping (causing the pressure to be reduced between the boiler and engine), and imperfect drainage, which is an enemy both to economy and to the life of the engine. In many of the newer plants it has been found a great advantage to install receivers to equalize the pressure and to collect the water before it reaches the engine.

In general it may be said that the principal cause for loss in steam-plants is the use of engines which are overloaded or unsuited to the conditions of work, undersized, or which have badly arranged steam- and exhaust-pipes. Other frequent causes are the imperfect condition and poor operation of the boilers. In many plants exhaust-steam, which might be utilized for heating, is wasted, and in others, where the exhaust-steam is utilized for heating, power is wasted by excessive back pressure. The most economical use that exhaust-steam can be put to is for heating, because thereby all the latent heat-units are made use of, but it should be done without back pressure on the engine,

by means of a vacuum system to draw the steam and water through the heating-pipes; otherwise there will be a loss both of fuel and of power, due to the engine working under imperfect conditions.

The system in general use for heating by exhaust-steam is by means of coils of pipe around the walls or overhead in factories and mills and by radiators in offices and private rooms.

If the space to be heated is not excessively large, the back pressure on the engine should not exceed 2 or 3 pounds per square inch, and may not exceed $\frac{1}{2}$ pound with the proper-sized exhaust-pipe and coil-connections.

The most serious trouble from back pressure in an exhaust heating-plant has been remedied by enlarging the area of the exhaust-pipe and of all the pipe-connections to the coils, so that the inlet of each coil is as large as or larger than the total area in each of the radiating-coils, resulting in the reduction of the back pressure from 4 pounds to $\frac{1}{2}$ pound per square inch.

CHAPTER XXIII

THE ENGINEER AND HIS DUTIES

WE cannot here enter into a description of the innumerable details and intricacies involved in the proper care of a steam-plant, such details and conditions having been very fully illustrated and described throughout the foregoing chapters of this work. The minor details and the knowledge required for operating a steam plant and for detecting or obviating the troubles and defects constantly arising therein are fully explained in the question-and-answer form in the many books on this subject, such as "Combustion of Coal" by Barr, and "Engine-Runners' Catechism" and "Steam-Engine Catechism" by Grimshaw—most interesting books for students and young engineers. Their study often affords hints useful to older heads, so that every engineer, on taking charge, should be thoroughly posted on the requirements of his profession in proportion to the extent of the duties assigned to him.

If all the duties devolve upon one person, as in small plants, the operating details, through all the steps from the coal-heaver to the expert steam-user, should be at his command, and if in charge of a large plant, where firemen, water-tenders, oilmen, cleaners, and machinists in repair-work are employed, his knowledge of the requirements of all detail work in the construction of the plant should not only be of the expert kind, but should also involve a large experience in the whole range of construction, its theory and practice, in steam-engineering. If an electrical-generating and transmission plant is in connection with a power-plant, he should be also an electrical expert that he may readily meet the contingencies that may occur in any part of a complex power-installation.

The license system is doing much for the education of engineers in the line of their duty, as its requirements impose an effort in study that would otherwise be neglected.

License is now required in the States of Massachusetts, New York, Ohio, Illinois, Wisconsin, Missouri, Minnesota, Kansas, Montana, and other States. Municipal license is required in many of the large cities.

KNOCKING AND OTHER NOISES IN THE ENGINE

The causes of knocking or other noises in a steam-engine are anxieties to the careful engineer from their numerous locations and signs of possible danger. They may be generally traced by the ear, or by the feeling of the hand or fingers in contact with different parts where possibly loose joints may occur, and in obscure cases by a small stick of hard wood placed between the suspected point and the fingers or teeth.

Some of these causes may be enumerated, commencing with the cylinder-head. Water in the cylinder gives a peculiar sound—a rush and a blow—while the contact of solid pieces gives a click or thump the character of which a little experience soon reveals. Looseness in the piston-rings; looseness in the follower; rattling of nuts, set-screws, or springs used to set out the packing-rings, by being cast adrift in the chambers of the spider section of the piston, produce a constant click at every stroke. Other causes of noise are looseness of the rod in the piston through faulty fastening; looseness of the end of the valve-rod in the valve-connection in the steam-chest or in any of its joints, direct or through a rocker-arm; looseness in the cross-head boxes and bearings, piston-rod key, or lock-nut. Oval bearings, bound brasses, and side-thrust should be examined, particularly the last-named at the crank-pin. Main journals on crank-end of shafts may wear and have a thrust-jar. Looseness in the side-bearings of the fly-wheel key and a loose joint in the made-up parts of a fly-wheel have sometimes been a mystery to find. Squeaking anywhere shows the want of oil.

DON'TS FOR ENGINEERS AND FIREMEN

Don't forget to look at the water-gauge or to try the gauge-cocks the first thing in the morning.

Don't forget to open the drip-cocks before opening the throttle, which should only be just started from its seat to allow the cylinder to warm up and discharge water.

Don't neglect to start the blow-off every morning, before pulling forward the banked fire, to clean out any sediment that may have

accumulated in the blow-off pipe from the use of muddy feed-water. Once a week will suffice for good water; and

Don't forget to blow off boilers, surface and bottom, if so arranged, at stated times, to suit the nature of the water in use.

Don't allow steam-traps on the cylinder or cylinders of one engine to be connected in any way to the steam-trap or discharge-pipe of any other engine, thereby causing water to be drawn back into a cylinder when the engine is stopped. Such neglect has caused a wrecked engine.

Don't forget to lift the safety-valve off its seat at least once every day, nor neglect to rig a lanyard from the end of the lever to a convenient place for this purpose.

Don't neglect to provide means for quickly closing the water-gauge valves when the glass breaks, if they are not automatic.

Don't forget your regular times for firing and for cleaning fires, and don't allow holes to burn in the fire-bed.

Don't let the ashes accumulate under the grate—choking burns the bars; have stated times for cleaning.

Don't forget to look at the water-gauge or to try the gauge-cocks often, and don't fail to regulate the running speed of the boiler-pump or injector to suit the requirement of an even water-level. A constant feed is best.

Don't forget to regulate the dampers and doors exactly to produce an even rate of combustion; if automatic dampers are in use, they should be often examined.

Don't neglect to blow out the steam- or water-gauge connection and also the pressure-gauge connection as often as needed to keep them free from obstruction.

Don't neglect to clean boilers at proper times to suit the kind of water used, by first drawing the fires, and, if brick-set, allowing sufficient time to cool the walls below damaging heat by opening the doors and dampers; then blow out and open the man- and hand-hole plates, and scrape out the scale and slush with a long hoe, and wash out with a strong stream of water from a hose.

Don't forget to see that the blow-off pipe is clear of obstruction after cleaning boiler by drawing water through it.

Don't neglect to clean the boiler-tubes as often as once a week, and in some cases twice or three times a week, according to the draught

of the chimney. A strong draught deposits less ashes in the tubes than a weak one.

Don't neglect to pump up your boiler to the upper gauge-line when stopping the engine at night. Start the pump before closing the throttle.

Don't forget to anticipate the stopping of the engine by throwing open the fire-doors, partly closing the draught-doors, and opening the dampers, and by spreading a little coal over the fire to prevent the sudden rising of the steam-pressure; then clean and bank fires.

Don't hang any old piece of iron on the safety-valve lever to stop sizzling. It's a dangerous practice. If the valve leaks regrind it.

Don't neglect to search for and find the cause of any unusual occurrence, noise, or knocking in the engine or boiler-pumps, nor put off the remedy to some more convenient time. To-day's doctor may prevent to-morrow's disaster.

QUESTIONS AND ANSWERS

The newly fledged engineer applying for a license cannot be expected to answer the thousand and one questions that may be contained in the catechisms of the examiners or inspectors, nor to understand the whys and wherefores of the elementary strength and construction of the machinery of the plant that he is to take charge of. It is sufficient if he has at hand the ready wit to operate and care for it and to know when it is running right or wrong, and what to do when confronted with the usual troubles of a power-plant. The progressive engineer has a vast field before him in which to explore the details of theoretical and constructive engineering that may lead him to the head of his profession.

We append a limited number of the leading questions and answers of vital interest to applicants; but in publishing them do not wish to depreciate the full study of the subject as shown in the published catechisms and text-books.

Question.—What is the most essential part of a steam-plant?

Answer.—The boiler, whose fire and water, by means of the heat of combustion, generate steam under pressure, which steam, by its expansive force in an engine, creates power.

Question.—What do you understand by combustion?

Answer.—Combustion is the production of heat by the union of the oxygen of the air with the carbon of the coal in the fire.

Question.—What is heat as you understand it?

Answer.—Heat is a property of matter as measured by its temperature, and the quantity of heat that matter can hold with its change of temperature.

Question.—Are there any other designations in regard to heat or its property?

Answer.—Yes; specific heat, which is the capacity of any body in units of heat to raise 1 pound of it 1° by the Fahrenheit scale; sensible heat, which is the measure of heat as indicated by the thermometer; and latent heat, which is the unit quantity of heat required to vaporize liquids or fuse solids per pound of their weight.

Question.—What is a unit of heat?

Answer.—A unit of heat is the standard of heat-measurement, and is equal to the quantity required to raise 1 pound of water 1° by the Fahrenheit scale, or from 39 to 40° F.

Question.—What are the essential requirements in the management of the fire under a boiler?

Answer.—A clean coal-bed and just enough air to produce perfect combustion.

Question.—What do you consider perfect combustion?

Answer.—The hottest condition of the fire, which requires 2 pounds of oxygen for the perfect combustion of 1 pound of coal.

Question.—How much air is required per pound of coal?

Answer.—As about one-quarter of the air is oxygen, it will require 10 pounds of air, or 130 cubic feet; but as the nitrogen of the air obstructs combustion, the best practice requires about 195 cubic feet of air per pound of coal fed to the furnace.

Question.—What is the effect of too much air fed to the fire?

Answer.—It has a cooling effect; as only the exact amount of its oxygen can be taken up by the coal to form carbonic-acid gas, any excess of air dilutes and cools the gases formed by combustion before they come in contact with the heating-surface of the boiler.

Question.—What is the effect if too little air is fed to the fire?

Answer.—The combustion is imperfect, and carbonic-oxide gas is formed of one-third of the heating-power, due to the coal, which becomes explosive by the admixture of fresh air.

Question.—What are the constituents of the gases from a boiler-furnace?

Answer.—Principally carbonic-acid gas (CO_2), carbonic oxide (CO), nitrogen (N), unconsumed oxygen and its nitrogen (excess of air), and steam from the moisture in the coal and air.

Question.—What effect has moisture or wet coal on combustion?

Answer.—They absorb heat by evaporation into steam, and retard the heat of combustion.

Question.—What are the safety-appliances usually attached to a boiler?

Answer.—Safety-valve, three gauge-cocks, water-gauge, pressure-gauge, and sometimes a draught-regulator, fusible plugs, and a low-water alarm.

Question.—How should a safety-valve be set?

Answer.—To blow off at or below the legal pressure allowed for the boiler. If a much lower pressure is used, 5 pounds above the usual requirement will be sufficient.

Question.—How should the water-gauge and gauge-cocks be set?

Answer.—So that the middle of the glass and the middle gauge-cock should be on a level with the proper water-level in the boiler—say from 4 to 6 inches above the tubes, according to the size of the boiler.

Question.—Where should the blow-off be attached to a boiler?

Answer.—At the back end, to the bottom of the back head, or to the shell for best effect; sometimes at the front head, with or without an extension-pipe reaching to the back of the boiler. A surface or scum blow-off is also desirable to discharge from the water-level.

Question.—What is a fusible plug, and what its use?

Answer.—A screw-thimble of hard brass, filled with pure Banca tin, which melts at 442° F., and usually screwed into the crown-sheet of locomotive-boilers to give an alarm by melting and blowing out when the water-level falls below the crown-sheet.

Question.—What is a steam-drum, and what its use?

Answer.—A reservoir from which usually to supply dry steam, but of doubtful value on boilers of full capacity for their work, as the drum weakens the shell.

Question.—What is a dry pipe, and what its use?

Answer.—A perforated pipe along the upper part of the steam-chamber of a boiler for distributing the area of the steam-inlet to the

steam-pipe, and thus preventing the priming or entrained water from entering.

Question.—What is an automatic damper, and what its use?

Answer.—A damper that is operated by the pressure in the boiler acting upon a regulator that opens and closes the damper, and thus controls the draught to equalize the boiler-pressure.

Question.—What is the effect upon the water-line of suddenly opening the throttle or the safety-valve?

Answer.—With small steam-room in the boiler and high pressure, the water would swell up by the liberation of steam, show a rise in the water-gauge, and probably carry water over to the engine, or discharge water from the safety-valve in case it was lifted.

Question.—What are the first requirements of an engineer or fireman when he enters the boiler-room in the morning?

Answer.—To try the gauge-cocks and open the water-gauge valves and the drip-valve to make sure of the water-level, and clear the gauge-glass connection. See that all valves are set properly as well as the damper; and if the fire has been banked, haul it forward and start it. Overhaul the pump and oil it; as soon as there is steam enough, start the pump running slowly; and do likewise with the injector, so that all may be ready at the time for starting the engine. See that all oil-cups have oil, and that all parts of the engine are ready to start; then open the throttle just enough to clear it and the pipes from water, the drip-cocks also being open, and warm up the engine under its slowest possible motion. Give it time—if a small one, one or two minutes, and if a large one, three to five minutes—to gradually get up to speed after the engine is warmed up and clear of water.

Question.—What is the effect of a surplus of air fed to a boiler-furnace?

Answer.—Air in excess of the amount necessary for perfect combustion tends to cool the furnace by abstracting heat from the gases of combustion.

Question.—What is the effect of feeding wet coal to the furnace?

Answer.—The water in the wet coal absorbs heat by evaporation, which does not produce combustion and the high temperature due to combustion, and therefore has a cooling effect upon the furnace.

Question.—In what direction is the steam-pressure in a boiler exerted?

Answer.—In all directions.

Question.—What part of a boiler has the greatest pressure?

Answer.—In the steam-space the pressure is equal in all directions; in the water-space the hydrostatic pressure of the water must be added to the steam-pressure.

Question.—How much is the hydrostatic pressure?

Answer.—It is equal to $\frac{43}{100}$ of a pound per square inch for every foot in depth.

Question.—If the upper valve on a water-gauge were closed, what would occur?

Answer.—The water would rise to the top of the gauge.

Question.—Why would the water rise?

Answer.—Because the steam above the water would cool, and its condensation would draw the water up.

Question.—What would be the effect of closing the lower valve only?

Answer.—The gauge would gradually fill up by condensation.

Question.—What would you do if you found the water out of sight in the water-gauge?

Answer.—Try the lower gauge-cock, then open the drip-cock to the water-gauge. If no water, stop the engine, throw open the fire-doors, damp the fire with ashes or coal, and feel the check-valve to find if the pump is feeding. If not, examine the pump, and if it has occasioned the trouble, start it running very slowly, when, if water appears in the water-gauge drip-cock, increase the pump-speed until water appears in the gauge-glass, whereupon regulate the fire and start the engine.

Question.—What would you do if the boiler commenced to foam excessively, or the water-gauge showed excessive motion?

Answer.—In ordinary cases increase the feed and blow-off to clear the water; if not found sufficient, check the fire, stop the engine, and prepare to clean the boiler.

Question.—What are the general causes for the foaming of boilers with good feed-water?

Answer.—The forcing of boilers that are too small for the work assigned them, dirty or greasy water, and boiler-cleaning compounds.

Question.—What would you do if you had a full head of steam and a good fire, and had to shut down suddenly?

Answer.—Open the fire-doors and cover the fire with ashes or coal, start or increase the pump-speed, and if the steam is still rising in pressure, lift the safety-valve.

Question.—What effect has the steam from a foaming boiler upon the engine?

Answer.—It carries water to the engine, which, by becoming solid in the pipe and steam-chest, is liable to wreck the engine by its solid filling of the clearance and compression-space. It is also wasteful of fuel.

Question.—Suppose that the pump was running and the water was going down in the boiler. What might be the cause and where would you look for it?

Answer.—Increase the speed of the pump and feel the check-valve to find if it is working, or try the test-cock on the force-pipe; find if the water-supply had failed; if there was no action of the pump, examine the pump-valves for faulty action; and, if necessary, stop the engine, slacken the fire, and overhaul the pump and water-supply; also examine the blow-off for leaks.

Question.—How would you find if the pump was not drawing water, or whether there was a stoppage in the suction-pipe?

Answer.—By rapping on the suction-pipe to find whether, by the sound, it is empty.

Question.—If you were feeding with an injector and it failed to feed, what would you do?

Answer.—Open the overflow and find if it were discharging. If not discharging, examine the water-supply. If steam were not discharging, open the injector and clean out the passages. Look after the boiler condition and its safety.

Question.—How often would you clean the tubes of a boiler?

Answer.—That would depend upon the kind of fuel and upon the chimney-draught; a strong draught tends to clear the tubes. With soft coal and weak draught, every other day with a steam-blower or brush; with anthracite coal, once a week is sometimes sufficient.

Question.—How often should a boiler be cleaned?

Answer.—That depends upon the kind of water used. With hard water, once in two weeks, with a daily blow-off; with soft, clear river water, once a month, with a blow-off every other day.

Question.—What is the difference between gauge-pressure and absolute pressure?

Answer.—Gauge-pressure is zero at atmospheric pressure, while absolute pressure starts from a perfect vacuum—14.7 pounds per square inch less than the mean atmospheric pressure.

Question.—What is the initial cylinder-pressure?

Answer.—It may ordinarily be the gauge-pressure in the cylinder at the beginning of the stroke, or, for the purposes of computation, the absolute pressure at that time.

Question.—What is back pressure?

Answer.—It is the retarding pressure on the piston during the stroke. In non-condensing engines it is that of the exhaust above atmospheric pressure, while in condensing-engines it is counted from a perfect vacuum.

Question.—What is the mean pressure in a cylinder?

Answer.—It is the mean forward pressure of the initial and expanding steam, less the mean back pressure from the exhaust above atmospheric pressure, or in absolute pressure above a vacuum.

Question.—What is clearance?

Answer.—It is the difference between the volume of the piston-displacement and the volume of the cylinder- and steam-passages, and varies from 2 to 8 per cent. of the piston-displacement in various types of engines. Its economy is inversely proportionate to its volume.

Question.—How can the loss by large clearance be modified?

Answer.—By early closing of the exhaust and causing compression to near the initial pressure.

Question.—Is the elimination of clearance possible?

Answer.—No; it is necessary in order to accommodate the lost motion in joints and prevent the piston striking the heads.

Question.—What is meant by working steam expansively?

Answer.—It is the cutting off the steam-inlet at some definite portion of the piston-stroke and completing the stroke by its expansive pressure.

Question.—What is the effect of using steam expansively?

Answer.—Its effect is in the economy due to the use of the expanding properties of steam below boiler-pressure.

Question.—To what extent could expansion be used economically in non-condensing engines?

Answer.—The economical nominal expansion can be carried to about one-twenty-fifth of the absolute steam-pressure.

Question.—To what extent for condensing-engines?

Answer.—About one-fourteenth of the absolute steam-pressure.

Question.—What effect has the clearance on the actual expansion?

Answer.—The clearance lessens the nominal expansion ratio, so that the actual expansion with clearance is less than the nominal expansion.

Question.—How early may a slide-valve cut off?

Answer.—About five-eighths of the stroke.

Question.—How can an earlier cut-off be obtained?

Answer.—By the addition of a riding cut-off valve, which may be adjusted for any desired cut-off.

Question.—How otherwise may a short cut-off be obtained?

Answer.—In a four-valve engine with one or two eccentrics, and in the Corliss type of engine.

Question.—How is the speed of slide-valve engines controlled?

Answer.—Generally by a flyball-governor operating a throttle-valve, or by a shaft-governor that varies the throw of the eccentric and of the valve.

Question.—What advantages has a riding cut-off on a slide-valve?

Answer.—It allows of any desired variation of the speed and power of the engine by a great range of the cut-off, and the full value of the steam used between its greatest range of pressure and temperature.

Question.—What advantages has the drop cut-off in the Corliss engine over the riding cut-off in other engines?

Answer.—It makes a more uniform admission-pressure, a sharper head to the expansion-curve, and a better control of the terminal exhaust- and compression-pressures. Its peculiar valve-gear allows of complete control of the movements of all the valves.

Question.—What is a condenser, and what its use?

Answer.—Any application of cold water for reducing steam to its primary condition of water and its use in the steam-engine is to save the value of its latent heat as a power-economy.

Question.—What are the principal types of condensers in use?

Answer.—The jet-condenser, in which a spray of water comes in contact with the exhaust-steam in a chamber; the surface-condenser,

in which the exhaust-steam is condensed on the surface of tubes made cold by circulating water; the siphon-condenser, in which the exhaust-steam is drawn into and condensed by a single jet of cold water under a hydrostatic vacuum made by a water-column about 34 feet high.

Question.—What advantage is a condenser to the power-economy of a steam-engine?

Answer.—It will add from 12 to a possible 14 pounds per square inch to the mean effective pressure above the atmospheric pressure.

Question.—What advantage has a surface-condenser over the jet- and siphon-condensers?

Answer.—It allows all the water of condensation to be used continually, taking the place of impure water. It is of especial value in the marine service and where good water is scarce.

Question.—Why are high steam-pressures advantageous?

Answer.—Because of the greater range of temperatures that can be utilized for power and their saving in steam by reduced cut-off.

Question.—What are the objections to the use of high pressures?

Answer.—They increase the danger of rupture at weak points in boilers and pipes, and of shock of moving parts, beside decomposition of lubricants, increase of leakage, and larger cost of the power-plant to meet increased pressure.

Question.—How is the economy of a steam-engine expressed?

Answer.—In the pounds of steam or its water consumed per hour per horse-power.

Question.—Why is it not expressed in pounds of coal?

Answer.—Because the boiler-duty is independent of the engine-duty, and the coal-duty should apply to both boiler and engine.

Question.—What is superheated steam?

Answer.—Steam is superheated at any temperature above that of the water from which it is generated, or above that of saturated steam.

Question.—What advantages are attributed to superheat?

Answer.—It lessens cylinder-condensation and its waste of power, and enables perfect expansion.

Question.—In what types of steam-engines is it most useful?

Answer.—In compound and multiple-expansion engines, where the loss of steam by condensation is greater than in non-condensing engines.

Question.—What may be the possible gain by superheating?

Answer.—The gain by superheat depends upon the method of obtaining it and its amount—in non-condensing engines, from 4 to 10 per cent.; in condensing-engines, from 6 to 14 per cent.; and in multi-expansion engines, from 10 to a possible 20 per cent.

Question.—What is the use of an indicator?

Answer.—To show the general conditions of the work of the steam by means of the form of the diagram that the recorder makes of pressures and volumes.

Question.—Does a well-proportioned diagram show an economical engine?

Answer.—Not always; leakages may balance each other and not affect the lines of the diagram.

Question.—What does an ill-proportioned diagram show?

Answer.—It shows where to look for the faults of the valve-motion and its construction, and also the degree of steam-economy.

Question.—What other important properties does the indicator-card show?

Answer.—A faultless diagram shows the indicated horse-power of the engine and the quantity of steam used per horse-power.

Question.—What is the use of a governor?

Answer.—To regulate the speed of the engine automatically, by varying the volume of steam inversely as the load.

Question.—What are the principles of action of governors?

Answer.—The throttling-governor varies the initial pressure, and the cut-off governor varies the volume of steam by varying the point of cut-off.

Question.—Which is the more efficient?

Answer.—The cut-off governor is much the more efficient.

Question.—What is a "stop motion" attachment to a governor?

Answer.—A device to stop the steam-supply in case the governor-belt breaks, or when the load is too greatly decreased, which over-speeds the engine.

Question.—What is lead on a steam-valve?

Answer.—Lead is the opening slightly of the steam- or exhaust-port before the crank reaches the centre.

Question.—How is lead obtained?

Answer.—By setting the eccentric ahead of the lap-angle; it is called the lead-angle on the eccentric.

Question.—What is lap?

Answer.—It is the extension of the face of the valve over the cylinder-ports both ways. Laps are designated as steam-lap and exhaust-lap.

Question.—What is the use of lap?

Answer.—To shorten the period of port-opening from greater valve-throw and its quicker motion.

Question.—How should the eccentric be set for the proper movement of a valve with lap?

Answer.—By advancing the eccentric, so that the steam-port just opens at the moment that the crank is on the centre, which is the lap-angle.

Question.—What effect has exhaust-lap?

Answer.—It increases compression by retarding exhaust-release and possibly choking it.

Question.—What effect has increase of steam-lap on the cut-off and expansion?

Answer.—It shortens the cut-off and prolongs expansion.

Question.—How early may a plain slide-valve cut off?

Answer.—About five-eighths stroke.

Question.—How short a cut-off may be obtained with a riding cut-off?

Answer.—From zero to five-eighths and three-fourths in various makes of engines.

Question.—What is the range of cut-off in Corliss engines?

Answer.—From zero to three-fourths.

Question.—What is a shifting eccentric?

Answer.—One that is moved from its centre to its extreme throw in a straight line by the varying centrifugal force of a shaft-governor.

Question.—What is a swinging eccentric?

Answer.—An eccentric with an arm pivoted to an arm of the fly-wheel, and swung in a circular arc from its centre to its extreme throw by the varying centrifugal force of a fly-wheel governor.

Question.—What are the advantages of high piston-speed?

Answer.—It lessens cylinder-condensation and enables greater power from lighter engines.

Question.—What are the disadvantages?

Answer.—Greater momentum and shock from the moving parts, causing increased wear and care in adjustment and lubrication.

Question.—What is the limit to fly-wheel speed?

Answer.—Practically according to the material and make-up of the wheel. Cast-iron wheels, solid, may have a rim-speed of about 6,000 feet per minute, although a much higher has been in use.

For the progressive young engineer there are a thousand or more questions which by their answers contribute to his advancement and success, and may finally put him at the head of his profession.

There is plenty of room at the head, and it only requires study and practice to reach it.

ELECTRICAL SECTION

PREFACE

The following pages have been prepared for the especial use of steam engineers, in order to give them, at a glance, the essence of modern electrical practice. A great many new departments of electrical engineering have crystallized in the last ten years, and in all probability more will assume commercial significance in the years to come.

For this reason, a very brief theoretical treatment has been given, in some instances sufficient, it is believed, to indicate the trend of future progress. The author knows that the two great problems facing the engineer in charge of a steam plant are: First, to keep his lights burning; and second, to keep his power supply going. In other words, the steam plant and the electrical plant in conjunction are used in central station and power-house work for lighting, as stated above, and for power purposes.

To sum the matter up still more explicitly, the power-house and central station, or any other case where steam and electricity are combined for a common purpose, are merely an instance of where electricity of a certain pressure and current is being produced. Its production, however, necessitates the knowledge and observance of certain laws, which absolutely govern the output of current. The efficiency of the plant is also a point of direct importance, likewise dependent upon the care and management demonstrated by those intrusted with such a responsibility.

In the following pages of text, in which the reader will find a variety of subjects treated, of direct interest to the engineer, the author believes information is being given in a terse and useful form for immediate use, especially in conjunction with the questions and answers appended.

A great deal of ground has been covered in these pages, and a great deal of information necessarily presented, with adequate explanations under the circumstances. The point kept in mind throughout by the author, however, was this: that the engineer is

a busy man, and prefers information and such principles as are by nature part of the facts, given in a simple and comprehensive manner. Lengthy explanations composing part of highly complex analyses would hardly be the thing in any case. Therefore, fundamental principles and what might be called fundamental facts compose the structure of the text. The troubles apt to arise in a piece of active machinery are of more direct importance to the steam engineer than involved theories. Some of the most important of these have been given consideration in connection with the parts of a dynamo or motor. Lighting in its various forms has also been reviewed and the salient and practical features of each given distinct attention.

On the whole, the engineer will be readily able to test the value of the information given, by utilizing it when the time comes. It is the author's belief that it will prove to be of great service, not only at a critical time, but as a means of presenting a bird's-eye view of what can be called modern electrical practice.

NEWTON HARRISON.

NOVEMBER, 1906.

CHAPTER XXIV

THE DYNAMO

OPERATION OF THE DYNAMO

THE operation of the dynamo may be best and most briefly described as that of the movement of conductors with respect to lines of force, or of lines of force with respect to conductors. By this is meant that in the ordinary type of direct- or alternating-current generator this idea (Fig. 1) predominates: namely, the movement or cutting of lines of magnetic force with respect to conductors, or the converse.

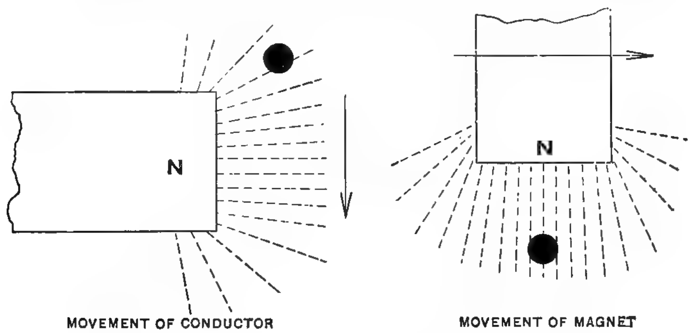


FIG. 1.—Effect of moving the conductor or the magnet.

It is evident from this statement that there are two kinds of generators: the direct and the alternating current. It is readily realized that differences must exist between one and the other type, distinguishing them in such a manner that they stand apart, as it were, representative of two systems. These differences, which are the means by which one kind of current is known from the other, are fundamental. The direct current is one which is unvarying or unchangeable (Fig. 2) in its direction. The alternating current is one which is constantly varying or changing its direction. The construction of generators, and to some extent their operation, are based upon the kind of current they generate, and the service that particular current performs.

The operation of the dynamo in general, therefore, represents either the movement of the conductors or the magnetic field relatively in

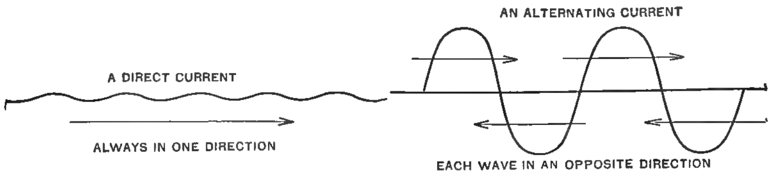
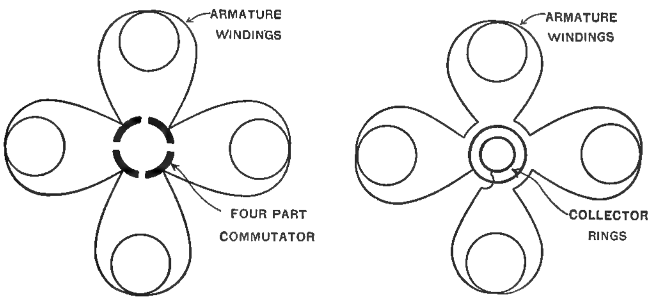


FIG. 2.—The tremor in a direct, and the reversals in an alternating current.

such a manner that by means of a commutator (Fig. 3) or collector-rings, or by dispensing with either, electricity is produced of the character of a direct or alternating current. As a general rule, in direct-



Use of commutator converts alternations into a direct current.

Use of collector-rings permits the natural alternations to pass outside.

FIG. 3.

current generators the source of magnetism or the field remains at rest. The armature with its conductors is set into rotation. In this case the conductors are made to cut the lines of force and generate electromotive force.

GENERATING ELECTROMOTIVE FORCE

A dynamo or motor is simply a generator of electromotive force. The electromotive force produced by the dynamo is utilized for lighting or power purposes. The electromotive force produced by a motor serves to regulate its current-supply, by acting automatically and oppositely to the pressure sending current in. The electromotive

force of a dynamo is calculated and developed with respect to the kind of work it is to perform. For instance, it will be 115 volts for incandescent lighting, or 500 volts for trolley-lines. The electromotive force is generated within the conductors when they cut the lines of force.

The basis of all theoretical and practical calculations is that a volt is produced if lines of force are cut by a conductor at the rate of 100,000,000 a second. When the elements of revolutions per second, conductors, and lines of force are considered together, they may be conveniently arranged as follows:

Volts = lines of force × revolutions per second × conductors on the armature ÷ 100,000,000.

Calling the volts E, the lines of force N, the revolutions per second S, and the conductors C, the formula may be written:

$$E = N \times S \times C \div 100{,}000{,}000.$$

For instance, if it is desirable to generate 115 volts, the field N may equal 5,000,000 lines of force (Fig. 4), the conductors C may

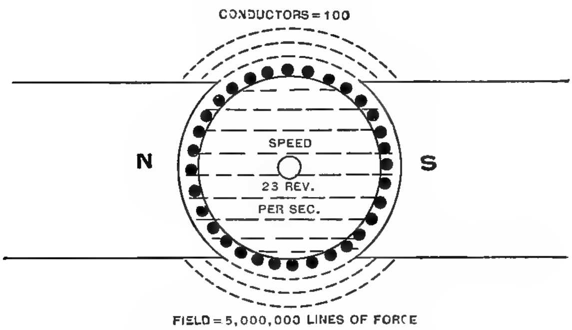


FIG. 4.—Field, speed, and conductors producing 115 volts.

equal 100, and the speed S may be 23 per second. On this basis the formula gives:

$$115 \text{ volts} = 5{,}000{,}000 \times 100 \times 23 \div 100{,}000{,}000.$$

All the different types of generators are constructed on this principle as a foundation. Whatever variations in appearance occur, they cannot be regarded as other than differences due to the various applications to which generators are put.

USE OF THE COMMUTATOR

Beginning with the generation of the electromotive force, it remains to be seen how the direction of the current is affected by it. It may be stated that the movement of a conductor past a magnetic north and south pole has the effect of not only generating electromotive force, but giving it direction, so to speak. By this is meant that a copper wire moved past a N pole will have a current pass through it in an opposite direction to a wire similarly moved past a S pole. Both may produce exactly the same electromotive force, but the direction of the current the electromotive force sets into action will be opposite in one case as compared with the other.

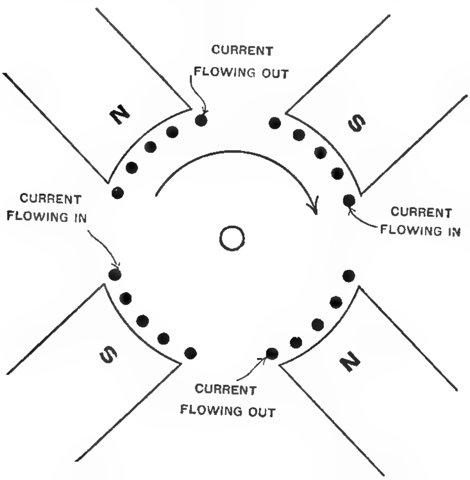


FIG. 5.—Currents under N poles flowing outward. Currents under S poles flowing inward.

A generator-field consists of two or more magnetic poles arranged alternately. The north and south poles follow each other respectively. Therefore a conductor will produce an electromotive force tending to send a current in opposite directions as it passes a N pole and then a S pole. There are many conductors on the armature of a direct-current generator. Conductors passing S poles will all carry a current in the same direction. Conductors passing N poles will all carry a current in an opposite direction (Fig. 5) to those passing the S poles.

The problem of successfully directing these two opposite but simultaneous flows of current into a circuit commonly called a direct-current circuit is solved by means of a commutator and brushes. The function therefore of a commutator is to conduct all electricity of one direction into one brush or set of brushes, and all electricity of an opposite direction into another brush or set of brushes. By this

means the naturally alternating current generated in the armature-conductors of a direct-current machine is rectified or commutated.

REGULATING THE DYNAMO

That which constitutes the regulation of a dynamo is accomplished by means of the field in the case of a direct-current incandescent-light shunt-wound machine. The field is either increased or decreased in strength, this being the means by which the electromotive force of the armature is raised or lowered. In other words, the fact that the armature rotates in a magnetic field of more or less lines of force is an evidence of the generation of a correspondingly higher or lower electromotive force. For instance, if 115 volts are obtained by means of 5,000,000 lines of force in the field, acting upon 100 armature-

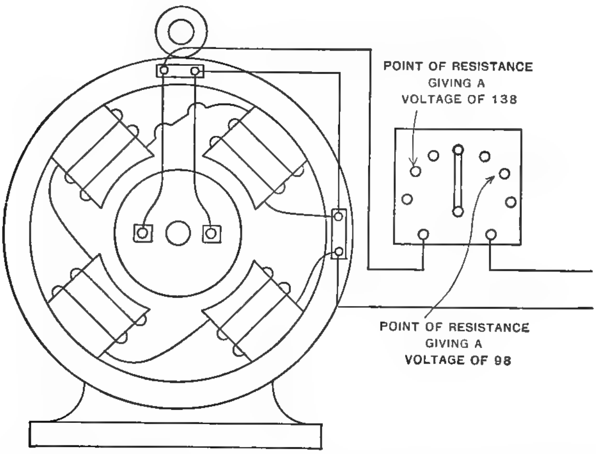


Fig. 6.—Effect of resistance in increasing and weakening the field strength and the electromotive force.

conductors rotating at a speed of 23 revolutions a second, an increase or decrease in the number of lines of force would mean in proportion just the same change in the volts generated.

If the 5,000,000 lines of force are increased to 6,000,000 or lowered to 4,000,000, the volts produced (Fig. 6) would be increased to 138 or lowered to 98. In other words, by causing a variation in the

strength of field by varying the current in the field-coils, a degree of effective regulation is obtained which has become established in common practice in connection with what are called shunt-wound dynamos.

CLASSIFICATION OF DYNAMOS

Dynamos are classified under two general headings: first, direct-current generators; secondly, alternating-current generators. The direct-current generators are further subdivided (Fig. 7) into series-wound machines, shunt-wound machines, and compound-wound machines. The alternating-current generators are classified with

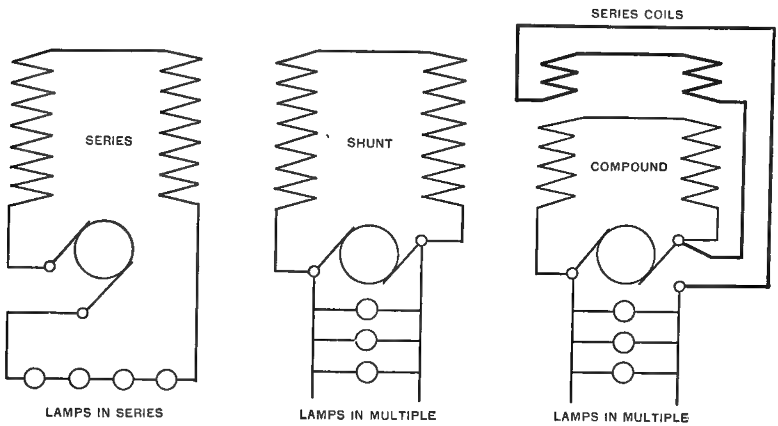


FIG. 7.—Three types of direct-current generators.

respect to the character of the current they develop. On this basis it may be said that there are single-phase machines, two-phase machines, and three-phase machines.

Among alternating-current generators are found forms of construction of a special character, in which neither the armature-wire nor field-wire is moved when electromotive force is being produced. An essential element of alternating-current practice is the transformer, by means of which the voltage is raised or lowered for transmitting or distributing the current.

REGULATION WITH A SERIES-WOUND DYNAMO

The series-wound dynamo is generally employed for a system of electric lighting in which a constant current is necessary, such as high-tension arc-lighting, for instance. The function of this type of generator is to provide a current of 10 or 12 amperes and a voltage that is capable of being adjusted by special means to suit the number of lamps in use. The arc-lamps are connected in series, each lamp taking the same number of volts. If twenty or thirty lamps are thus connected, twenty or thirty times the volts required for one lamp is necessary. Allowing 50 volts as the amount required for each lamp, the total voltage to be generated would equal 20×50 or 30×50, or from 1,000 to 1,500 volts.

Arc-lamps as now used may be of the open or closed arc type. By this is meant that the carbons either burn in the open air, lasting only about eight or ten hours, or are enclosed in a small globe. Each lamp takes 50 volts if of the open air type. If of the closed globe type each lamp will take about 80 volts. The dynamo must be able to automatically raise or lower its voltage when lamps are turned on or off. When more lamps are added to the line, more volts will be required; in fact, as much more as there are extra lamps.

When lamps are cut out, less volts will be required, in proportion to the number of lamps. For instance, if ten lamps of the closed globe type are added to the circuit of a series-wound dynamo, by simply turning them on, 10×80, or 800 volts more must be sent into the line. On the other hand, if ten lamps are cut out, 800 volts less in the line will do. Fewer lamps turned on or off would mean the same thing—more or less volts accordingly. The series-wound dynamo, with an automatically varying voltage as described, but a constant current, meets these requirements by shifting its brushes (Fig. 8) automatically around its commutator. This is based upon the principle that between two given points on the commutator the generator gives out its highest and its lowest voltage. If the brushes can be made, by means of the attraction of an electromagnet, to move to such points of the commutator that these points will supply just the volts required by the number of lamps in use, the problem is evidently solved.

The actuating magnet which causes this regulation is also in series with the line, and is sensitive to the current in the line. If for an instant that current is too high, the power of the magnet is sufficiently increased to move the brushes over to a point where the pressure drops. If the current in the line is too weak, the power of the magnet reduces sufficiently to permit the brushes to assume a new position

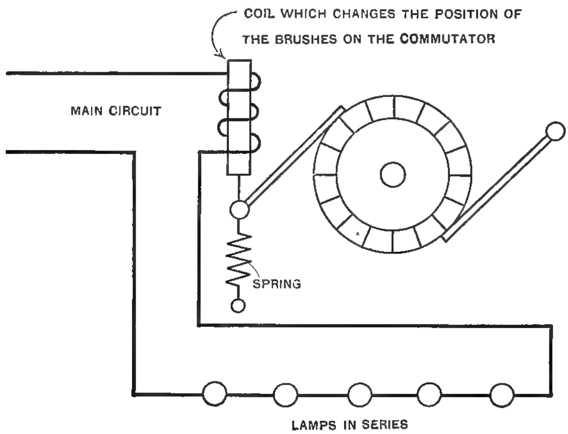


FIG. 8.—Method of shifting the brushes.

on the commutator, where more volts may be sent out over the line. This adjustment is continually going on in a series arc-light machine of modern construction. Though other methods of regulating the voltage and keeping the current constant have been tried, this method has been generally accepted and adopted as the best to be used under the circumstances involved.

REGULATION WITH A SHUNT-WOUND DYNAMO

The shunt-wound dynamo is one in which the fields take but a small percentage of the total current. In large generators the fields take from 1 to 2 per cent., or even less. In small generators the field-current will represent a heavier percentage of the total armature-current. In practice a resistance is inserted in the field-circuit, so that its manipulation will control the amount of current passing into the field-winding. When a shunt dynamo is being loaded up, the ten-

dency of the lamps to drop in candle-power becomes more and more apparent.

The causes operating to bring about this effect are found (Fig. 9) in the machine as follows: First, drop in the armature-conductors due to the armature-current passing through the armature-resistance, represented by C×R; secondly, demagnetizing-field of the armature, through which the influence of the armature as an electro-magnet opposing the field proper becomes more and more magnified. These disturbing influences may be readily understood as acting as means of cutting down the volts obtainable from the generator to a marked degree, unless an effective remedy is employed.

It is evident that the volts lost in the armature, through drop of potential, can only be compensated for by generating that number of volts extra. It is also evident that the reactive magnetic effect of the armature can only be compensated for by providing as many more lines of force, or as much more magnetism, as has been rendered ineffective by it. In other words, the only way to regain the volts actually lost in the armature, and those additional volts which

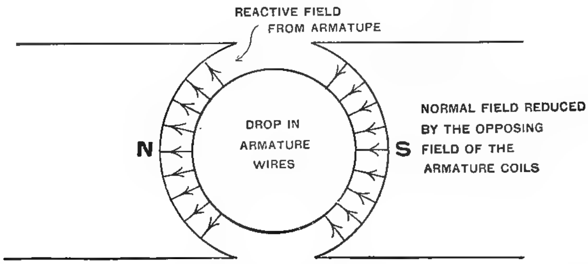


Fig. 9.—Armature-reaction and armature-drop in a shunt-wound dynamo.

would be generated in it were some of the power of the field not destroyed, is by supplying enough extra lines of force when required to meet this emergency.

For this reason, therefore, the rheostat inserted in the field-circuit is so utilized that when the dynamo is running with a light load, little or no current passes through it. When the load is increased, the rheostat is so adjusted that the field-windings take more current. Cognizance of the candle-power of what is called the pilot-lamp, or of the pressure indicated by the voltmeter, will enable the attendant

to move the rheostat to the proper point. This is not an automatic system of regulation, because of the constant observation required of the attendant. Automatic preservation of the normal or working-lamp pressure is obtained by means of the compound-wound generator.

REGULATION WITH A COMPOUND-WOUND DYNAMO

Preserving the pressure automatically by means of a compound winding, simply means the employment of a winding, or rather of two windings, one of which is simply a shunt winding, and the other a series winding. In other words, the advantages of both are combined in such a manner that when the generator is called upon for more current, it provides it without any external signs of a drop of pressure.

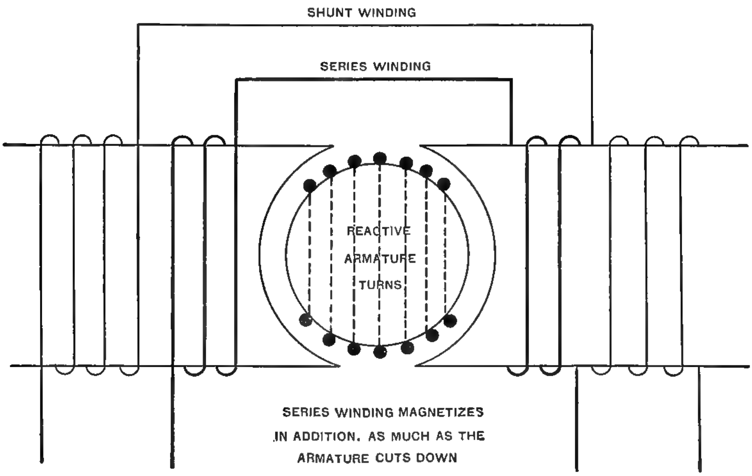


FIG. 10.—Series winding produces lines of force that equal those the armature destroys.

Through this means, at a low point of load, the lamps are not too bright, or at a high point of load too dim. The general principle involved is that of sending through a few turns of heavy wire (Fig. 10) wound around the field all the current the dynamo generates. In this manner the more current the dynamo produces the stronger it makes its field in consequence.

When there is little or no current coming from the armature,

the field is only that produced by the shunt winding. Where there is a heavy current produced by the armature, not only is the shunt winding supplying its quota of magnetism to the field, but its effect is augmented to the extent of the magnetism supplied by the few turns carrying the total current, commonly called the series-turns, in contradistinction to the shunt-turns. Magnetism is therefore produced in the compound-wound machine by two sets of coils whose effects are coöperative. The shunt winding in such a case does not differ in character or principle from that of an ordinary shunt-machine. The series or compensating winding is there for the purpose of adding as much additional magnetism to the field as is required to compensate for the armature-reaction, and of supplying as many extra volts to be generated as will make up for those lost by drop.

For instance, if the shunt-field supplies 5,000,000 lines of force, and the armature at no load gives 115 volts, the machine at full load, without the series-coil, may have an effective field of only 4,000,000 lines of force and give only 95 volts. This would mean very bad lighting and a great waste of power. The series-coil therefore adds enough extra magnetism to build the field up to, and keep it constant at, a little over 5,000,000 lines of force. A little over 5,000,000 lines of force are necessary, because there is armature-drop as well as field-reaction to compensate for. Thus, in a case where the volts drop at full load, from the initial pressure of 115 to 95, the weakened field may account for 15 or 18 volts, and the armature-drop for about 2. There is a little drop as well in the series winding itself to allow for, and the switchboard and the heavy mains leading thereto are not to be forgotten in including all possible sources of loss of pressure in the machine and the points of distribution of its power.

CHAPTER XXV

Testing

TESTING A DYNAMO FOR FAULTS

A DYNAMO may be tested for faults such as are usually denominated grounds, short circuits, sparking, or failure to generate, etc. It is best to treat of these conditions categorically, in order that each may appear in its true aspect. If the field-coils are grounded, due to moisture, poor insulation, or the actual contact of copper to iron, two cases are presented: first, a ground in one coil; secondly, a ground in both coils. Heat is the phenomenon which always presents itself in one or more coils due to these causes. If one coil becomes heated to a much greater extent than the other, the cool coil is either severely grounded or short-circuited.

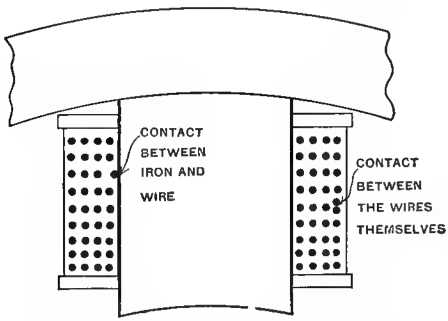


FIG. 11.—Wire touching iron, and wire touching wire.

The ground may have occurred at either the beginning or the end of the coil. If at the beginning, the coil will be cooler than if it had occurred at the end. By this is meant (Fig. 11) that if sufficient length of the coil carries the current, it will be warmer than if it only passed through a short length and then entered the other coil. The cooler coil, on the other hand, may be in contact with the iron or itself at two places. In such a case it will be short-circuited, by which is meant that a greater or a lesser part of the winding is cut out.

If a greater or a lesser part of the winding of one coil is not in circuit, the current does not meet with the same resistance, but with less, and in consequence the second coil is carrying too much current

as well as part of the first coil. For two coil-fields this is true, though the same principle of locating the cooler coil or coils, where more than two coils are in series, will enable the attendant to draw conclusions as to the ground or short circuit for purposes of repair. Baking a coil to dispel moisture is often a means of eradicating latent faults of this character.

SPARKING

Sparking may be caused by too great a load on the dynamo, or by a wrong position of the brushes. One brush may not be diametrically opposite the other in a two-pole machine, or the brushes may not be properly adjusted on the commutator if they belong to a multipolar machine. If a four-pole machine, they should be 90 degrees apart, if a six-pole machine, 60 degrees apart, etc. The simplest method of getting the correct distance between brushes is to count the commutator-bars and divide them numerically by the number of poles of the generator. Sometimes the cause of sparking is mechanical, by which is meant that the bars may be loose, or either the bars or the mica, or both, project beyond the commutator in general. The difficulty is frequently found (Fig. 12) in the more rapid wearing away of the copper bars before the mica itself has worn down.

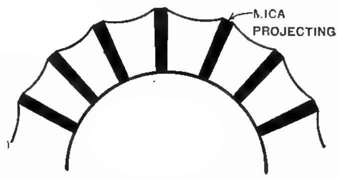


FIG. 12.—Mica wearing away slower than the bar.

The mica is a mineral product of a greater hardness than might have been expected, and in consequence the friction of the brushes affects it the least. Sandpapering the commutator is of little or no use. The only effective remedy is that of turning down the commutator—a thing best done by a lathe or by a special commutator turning device found on the market. The brushes may not be of the proper quality, and thus cause sparking. For instance, as a matter of information it may be stated that carbon brushes will not do in all cases. Carbon brushes are best suited to hard-drawn copper bars, not to soft ones. Segments of a less durable material will only be ground away in a fine powder, whose ultimate injury to the machine will be greater the longer this condition exists. In addition it may be stated that hard or soft carbon brushes may work with different

degrees of efficacy, according to the type of machine and the nature of the design.

There is such a thing as ineradicable sparking, due to bad design. In such a case as this little can be done by those intrusted with the operation of the machine. Sparking is sometimes caused by a too small air-gap between the armature and the field. As previously stated, it may be due to a too great overload of the machine. There are features involved in the design of a sparkless machine which bring into close relation the arc of embrace of the pole-piece, the length of the air-gap, the ampere-stream under each pole, and the magnetic spray from the pole-piece. The position of the brushes, which gives what is called sparkless commutation, is largely controlled by the existence of a magnetic fringe which extends beyond the leading pole-tip.

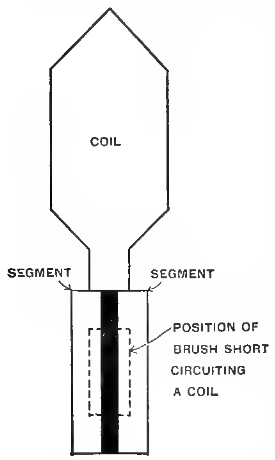


FIG. 13.—Time at which pole-piece spray is useful.

The state of this fringe indicates, in certain respects, a measure of the amount of sparklessness possible to achieve. The adjustment of the brushes is made for the purpose of securing a point on the commutator where the reversal of current in the successive armature-coils passing under the brush or brushes is accomplished without sparking. This magnetic fringe or spray is effective in reducing inductive effects in the coil or coils undergoing reversal, by introducing an opposite influence. Therefore the bad effects of reversal when the brush short-circuits the coil (Fig. 13) and the influence of the fringe counteract each other if the design of the machine is correct in these details. Sparking is therefore generally inherent, though sometimes due to the causes noted under the head of mechanical defects, or to a lack of judgment in selecting a suitable brush for the generator.

DYNAMO FAILS TO GENERATE

The development of electromotive force is the primary and essential feature of a dynamo's operation. If the machine will not "pick up," that is to say, begin to generate its electromotive force, there

are certain possibilities causing this condition, which may be enumerated as follows: The residual magnetism may be absent, in which case it must be restored by means of a current sent into the field-coils in the right direction. The magnetic poles of the machine may be the same (Fig. 14), in which case the connections must be changed. The field-coils may have an open circuit, in which case they must be tested, and the coil or coils repaired.

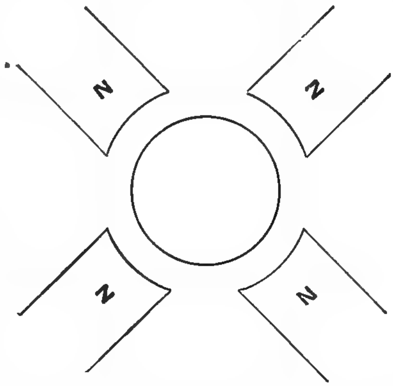


FIG. 14.—Case in which the poles are alike.

A more scientific reason than these may account for the inability of a dynamo to generate electromotive force by referring to its shunt-resistance and speed. In other words, the simple fact of the matter is this: that if the speed of the armature is not suited to the resistance of the shunt, due to this said resistance being too great, the machine will not generate. There is what is called a critical speed for shunt- or compound-wound dynamos, below or above which normal conditions will not exist.

If the generator or its outgoing circuits have made a double ground, the equivalent of a short circuit, the dynamo will not generate. A test of the circuits is therefore a necessary prelude to investigations of this character. Faults in the armature itself may be found in the nature of short circuits of one or more coils, which coils will get very hot on running the machine, this condition preventing any considerable outside pressure from appearing. A broken coil, on the other hand, causes sparking and the blackening of the bars between the ends of the break.

CAUSE OF HEAT IN THE ARMATURE

The existence of heat in the armature may be due to a variety of contributing influences. The dynamo may be overloaded, or the heat may issue or be conducted from some other source than the

armature itself. There may be difficulties present in the coils, such as, for instance, short circuits. The armature may be filled with dampness and generally grounded, a condition removed only by means of a baking process. Certain coils may be so wound that they do not send their currents in the proper direction. If the winding is reversed, a form of parasitical current of large amperage develops, causing great heat.

A source of disturbance and heat is discovered in the use of very thick copper wires or bars. The effect of these is a little extraordinary in the sense that they may develop currents within themselves strictly parasitical in nature. The thick bars may have eddies of electricity (Fig. 15) flowing in them, of such strength that they become intensely hot. These eddies are due to the fact that one-half of the bar longitudinally is developing electromotive force, while the other half is not.

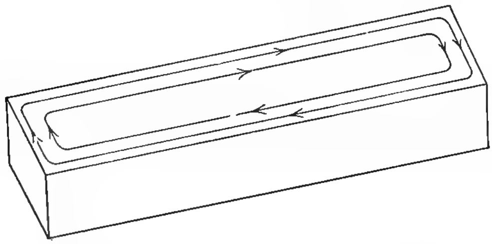


Fig. 15.—A copper bar with parasitical currents.

They may occur when one-half the bar is affected by lines of force as it is entering under a pole-edge a little sooner than the other half, simply on account of its extreme width or thickness. The brushes may be in the wrong position, on the other hand, and give rise to heating, or two commutator-bars or a commutator-bar and the frame or bushing may touch together. In running a generator the signs of excessive heating must be carefully watched for: otherwise the fact will be heralded in the way of a smell of burning insulation at a time when the need of light and power is of vital consequence.

HEAT IN THE COMMUTATOR AND BRUSHES

Too much friction (Fig. 16) between the commutator and brushes is a prolific cause of heat. Yet the contrary is true, that when the contact between them is insufficient, heat is developed due to the passage of the current through a comparatively high resistance. The heat generated by means of an electric current is measured in watts by squaring the current and multiplying by the resistance.

For instance, if 10 amperes are passing through a resistance of 10 ohms, according to the law governing such cases, the watts wasted will equal 10×10, the current squared, multiplied by 10, the ohms' resistance, or $100 \times 10 = 1{,}000$ watts. The heating effect is therefore measurable by the waste of watts on this basis. And the heat developed may be also noted to be proportional to the square of the current in amperes.

By this is meant that if the 10 amperes are doubled, the watts will be quadrupled. If the 10 amperes are made 20, the watts wasted become $20 \times 20 \times 10 = 4{,}000$ watts instead of 1,000. In other words, the effect of twice the current, from the heat standpoint, is to multiply the heat four times. The effect of three times the current would be to multiply the heat nine times, if the resistance remains the same.

Where commutator and brushes are concerned or any conducting part carrying a very large current, the least resistance may mean a very large amount of wasted power. The brush may press

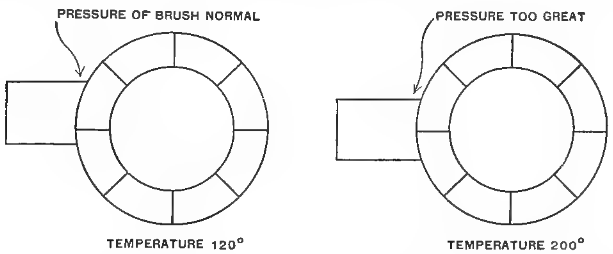


FIG. 16.—Temperature of commutator affected by brush pressure

properly, but may have too great a resistance and develop heat, or the casing holding it may not make the proper contact and cause heat to manifest itself. An ordinarily good contact will prove to be very deficient in any case where a very heavy current passes through it, as just indicated.

RADIATING SURFACE OF COILS AND CURRENT-CARRYING PARTS

In order to present with adequacy the subject of heat in the various parts of an electrical machine, it is necessary to touch upon the physical or geometrical facts (Fig. 17) concerning the rise of temperature

in conductors, coils, armatures, commutators, and brushes. As far as the geometrical facts are concerned, it may be said that the larger the surface of a heated body the more rapidly it cools. The converse is also true, that the smaller the surface of a heated body the hotter it becomes. From the standpoint of physics, however, temperature is merely an indication, so to speak, of the degree of concentration of the heat.

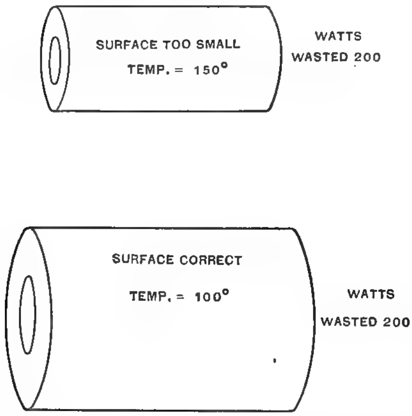


FIG. 17.—Effect of radiating surface on temperature with the same waste of energy.

By this is meant that with a given quantity of heat, say that given out by a candle-flame for an hour, a mass of material with a large surface would not show a high temperature. A small mass of material, however, would show a very high temperature, particularly if its surface was very small. From this standpoint, therefore, temperature is dependent upon not only the amount of heat actually present, but the rate at which it is being radiated. In this sense a conductor carrying electricity and developing heat will rise in temperature according to the amount of metal it represents in proportion to its outer radiating surface.

Coils carrying electrical energy, such as field- or armature-windings, or commutators or brushes, must be provided with sufficient surface for radiation to get rid of the heat quickly enough to prevent any but a limited rise of temperature. In steam-engineering the general problem is such that it is a matter of economy to retain the heat as far as possible. In electrical engineering the process is generally reversed. Getting rid of the heat as quickly as possible is a feature of daily practice.

TYPES OF MOTORS IN SERVICE

The types of motors in service are best classified under the titles of series, shunt, and compound or differentially wound machines. The series motor is one in which the speed rises to a destructive point

without a load or curb. The shunt motor is one in which it is necessary to have a resistance in series with the armature when starting it up. The compound or differentially wound motor is one in which the series winding acts either to increase the speed of the motor, or to increase its pull or torque with a heavy load. The reason why the series winding acts this way is simply because with one method of connecting its terminals the field is cut down. By reversing the connections (Fig. 18) the field is built up. In a shunt motor the weakening of the field means a higher speed. The strengthening of the field means a greater pull and a lower speed.

Armature-coils may be burned out or grounded, or the field may be deficient in the motors whose types are given. A general principle of great value in relation to the speed and pull, with respect to the practical application of direct-current motors, is that the field-strength and armature-current are the dominating factors. As a general rule, the greater the armature-current and the strength of field, the greater the pulling power of the machine.

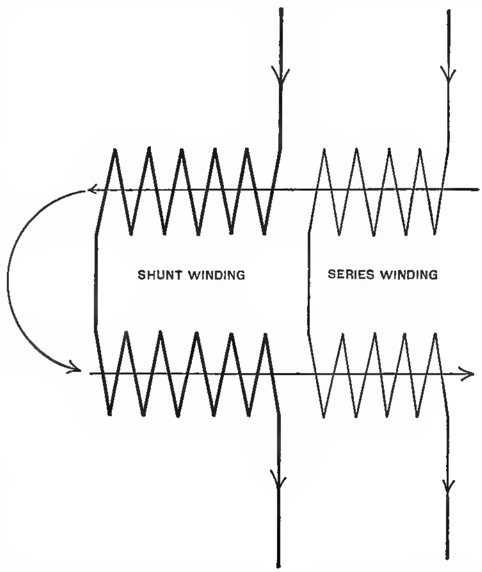


Fig. 18.—Both series and shunt winding acting to increase the strength of the field.

There is such a condition inviting danger to the motor as too low a speed. A shunt-, series-, or compound-wound motor too heavily loaded will naturally carry too much current, heat too much, run too slow, and probably spark too much in service. A shunt- or compound-wound motor running too fast is a case (Fig. 19) in which the field is dangerously weak. The shunt-coils in such a case should be carefully examined, and the series-coils reversed in connections.

A principle enunciated by Jacobi many years ago relates to the

maximum torque of the armature in about the following terms: A motor is doing its maximum work when it is loaded to such a point that its speed has become one-half the normal value. In other words, the heavy loading of a motor may be effective in producing a surplus of power, but this is only done at the sacrifice of efficiency. The over-

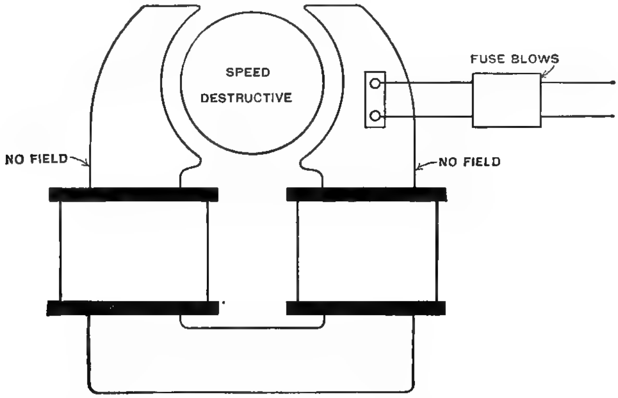


FIG. 19.—A shunt- or compound-wound motor with no field.

loading of a motor therefore to the point noted, namely, half the speed, means a theoretical efficiency of only 50 per cent. and a practical efficiency of even less. The average motor-efficiency cannot be given in exact figures except for a given size and type, but it may be stated on good authority that it is over 80 per cent. and less than 95 per cent.

As regards the armature-coils being burned out or grounded, or the field-coils exhibiting deficiencies, it may be said that the first case will be manifest in the shape of an unusually heavy current with probable fuse-blowing or circuit-breaker action; the second will be in evidence in the form of a weak magnetic field with phenomena as noted in the way of high speed above the usual value.

SPARKING IN THE MOTOR

The presence of a very high speed, and a line of sparks around the commutator, with two of the bars deteriorated and burned, means a case of open circuit in the armature-winding. When one of the coils or both are out of order, so that little or no field is present, the armature will spark badly if turned by hand with resistance in circuit.

The field-windings may be opposed to each other, and in this case the armature will tear itself to pieces if run idle without a preliminary examination of some sort. The rules which seem to be best to observe in connection with shunt- and compound-wound motors is to see that the starting resistance is in series with the armature when it is started, and that the field is on before current enters the armature.

THE BACK ELECTROMOTIVE FORCE OF A MOTOR

The back electromotive force of a motor is best understood as due to the rotation of the armature-conductors in the magnetic field. Through this, electromotive force is generated which has a polarity opposed to that of the entering pressure. To distinguish one from the other, the line-pressure is called the impressed electromotive force (Fig. 20), and the armature-pressure the back or counter electromotive force. The exact value of the back electromotive force may be calculated by means of the current and the armature-resistance.

If the current in the armature at any point of load is multiplied by its resistance and the product subtracted from the impressed electromotive force, the difference is the back electromotive force.

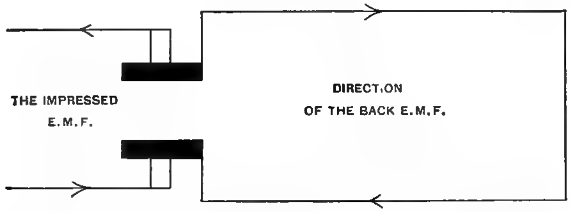


FIG. 20.—Opposition of the back to the impressed E. M. F. in a motor.

For instance, if the armature-resistance is .1 of an ohm, and the armature-current 50 amperes, the product is $.1 \times 50 = 5$ volts; subtracting this from an impressed electromotive force of 250 volts would give a back electromotive force of $250 - 5 = 245$ volts.

Electrical efficiency is obtained by dividing the 245 volts, back electromotive force, by the 250 volts, impressed electromotive force, or $245 \div 250 = 98$ per cent. But the electrical efficiency is not the one distinguished by the title "commercial efficiency." This last efficiency is equal to the ratio between the output and the input; or,

in other words, it may be stated that the commercial efficiency is equal to the power taken out divided by the power sent in.

HUMMING AND OTHER NOISES IN MOTORS

Humming and other noises are associated to a very marked extent with motors in operation. What is commonly called humming is found in motors in which the armature-slots carrying the conductors are not properly related to the pole-tips of the machine. There is a certain snap when the armature-tooth (Fig. 21) leaves the pole-tip. The effect of this magnetic snap is to set up a molecular vibration in the pole, which is sometimes greatly augmented by favorable acoustic conditions. The other sounds may be regarded as due to a vibration of the brushes with respect to the commutator of the motor.

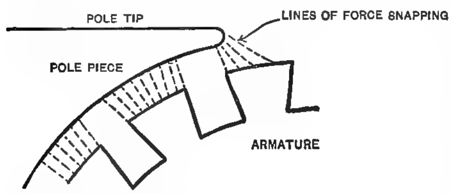


FIG. 21.—Cause of hum between pole-tip and slotted armature.

A peculiar chattering, as it is called in the machine-shop, is due to the lack of proper inclination of the brush or brushes to the commutator. The commutator, on the other hand, may be rough and require turning down. A method of localizing the sound is to lift the brushes off the machine when running idle. In this manner it is possible to ascertain whether the noise is an accentuated hum due to the armature-teeth or the commutator and brushes. A little oil or vaseline is frequently efficacious in this respect. It is good practice to file the brush carefully to the proper angle to remedy this evil.

Another salient cause of vibration and noise, however, is the lack of balance in the armature itself. It is a common practice, after a motor or dynamo-armature has been completed, to test it for mechanical balance on knife-edges. It is sometimes done so hastily or carelessly that the balance is imperfect. This can be readily discovered in the running machine by means of the hand when placed upon it. There is always a possibility, however, that the pulley may be defective. In this case it should be removed and carefully turned down until balanced. Removal of such disturbances better insures the period of usefulness of the machine.

CHAPTER XXVI

THE SWITCHBOARD

THAT which is typified by the name of switchboard in connection with central stations, power-houses, or private installations, is simply a convenient centre from which or to which all important conductors are led (Fig. 22), and at which the instruments and protective apparatus may be found. The switchboard is generally made of slate or marble, and of sufficient size to contain on its polished surface not only the terminals of circuits, but a variety of unique but nevertheless indispensable adjuncts of the equipment. It is necessary to classify the elements constituting the direct-current switchboard

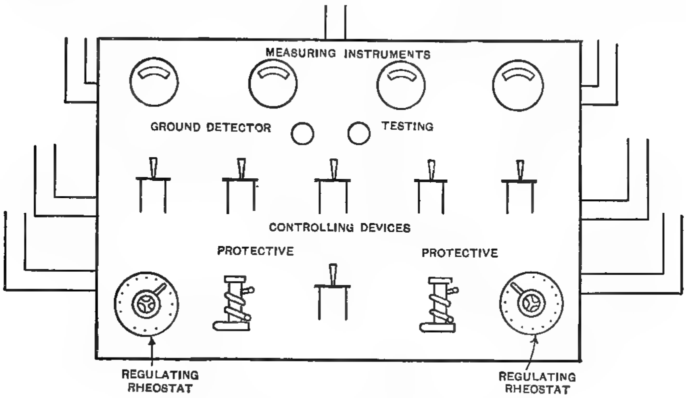


FIG. 22.—Circuits connecting to switchboard.

equipment in order to form an adequate idea of its importance and serviceability. This classification would assume the following form:

1. Measuring-instruments, by which would be included those employed for the measurement of volts, amperes, and watts.
2. Controlling devices, by which would be included all switches controlling main, feeder, and subsidiary circuits.
3. Protective devices, by which would be included all fuses, cut-

outs, and circuit-breakers of all kinds or shapes and embracing within that scope lightning-arresters as well.

4. Regulating devices, under which heading the rheostats in the fields or shunt-circuits and the bus-bars would be consistently included for this purpose.

5. Testing devices, under which heading would be included ground-detectors and such additional instruments as may be used to serve the same purpose, and may therefore be regarded as switchboard accessories.

The classification of the wires is part of the same proposition, for it is evident that the original purpose of the switchboard was only to coördinate or centralize the most important wires. For this reason the circuits may be regarded as belonging to one or the other of such divisions as the entire classification of the switchboard may include.

CLASSIFICATION OF CIRCUITS

The natural arrangement of conductors would be in the order of their essentiality or importance. For this reason the wires from the dynamo come first and are called mains. The wires from the mains are generally employed as supply-wires or feeders (Fig. 23) to the dis-

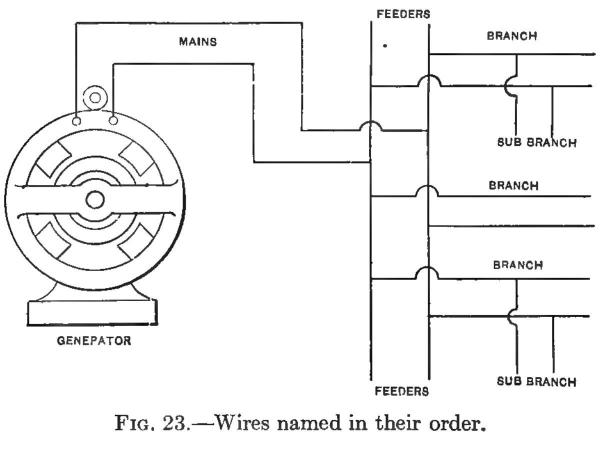


Fig. 23.—Wires named in their order.

tributing-wires with which they are connected. The feeder-wires supply current to the branches, and by means of the more important or heavier branch wires connect a class of sub-branches or subsidiary

circuits. On this basis there appear (1) mains, or wires from the generator; (2) feeders, or wires from the switchboard; (3) branches of a heavier character supplied by the feeders; and (4) sub-branches, or final distributing-wires.

CENTRES OF DISTRIBUTION

The switchboard carries the devices which exercise the various influences noted over the whole system. There are, however, points on each floor, or group of floors or rooms, in which a secondary influence may be exercised. Such points are called " centres of dis-

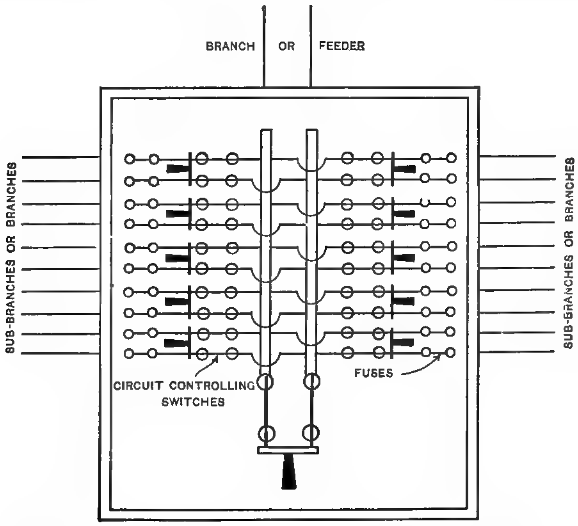


FIG. 24.—Use of a panel board at a center of distribution.

tribution." They appear in the form of miniature switchboards (Fig. 24) which are devoid of other than controlling and protective appliances, namely, switches and fuses.

The feeders enter these panel boards, as they are called, and from them various branches radiate to the groups of lights receiving the current. All important branches connect with the feeder through the medium of a switch and fuse. The sub-branches may or may not be provided with these, depending upon the character of the lighting; but the entire system of wires on this basis, from the generator to the lights, is adequately protected against overflows of current.

SWITCHBOARD APPLIANCES

The switchboard, as an entity, is itself subdivided, according to the purpose its different sections serve. These sections are now made in distinct panels (Fig. 25) or part-switchboards, according to the following system:

(1) The generating-panel, with which the generator connects directly with its controlling and protective devices. (2) The metering- or load-panel, with which the voltmeter, ammeter, and wattmeter connect. (3) The feeder-panel, with which the outgoing feeders, with their controlling-switches, connect, and which might be called the distributing-panel. (4) The testing-panel, to which the ground-detectors, lightning-arresters, etc., may be connected.

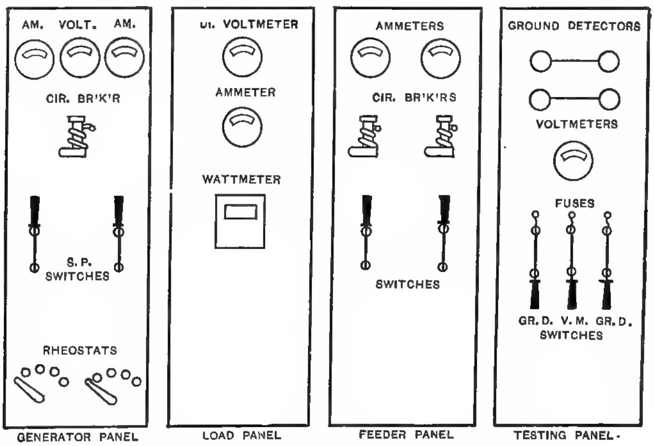


Fig. 25.—Elements of a switchboard.

The ammeter or ampere-meter is an instrument through which the entire current of the circuit to which it is connected passes. By indicating the amperes it practically records the number of lamps in service, unless motors are also feeding from the line. This instrument is always placed in *series* in the line, not in multiple; if placed otherwise it would be destroyed or greatly disabled. The voltmeter is always placed in *multiple* with the circuit (Fig. 26) whose pressure it indicates. The reason for the difference in connections of the am-

meter and voltmeter is found in the great difference in the resistance they represent.

The ammeter has the least possible resistance; the voltmeter, on the contrary, the highest possible resistance. The ammeter would cause a short circuit to the line if placed across its terminals; the voltmeter, if placed in series, would completely block the passage of the current.

The wattmeter indicates the product of the two elements of power noted by the ammeter and voltmeter (watts = amperes × volts), and

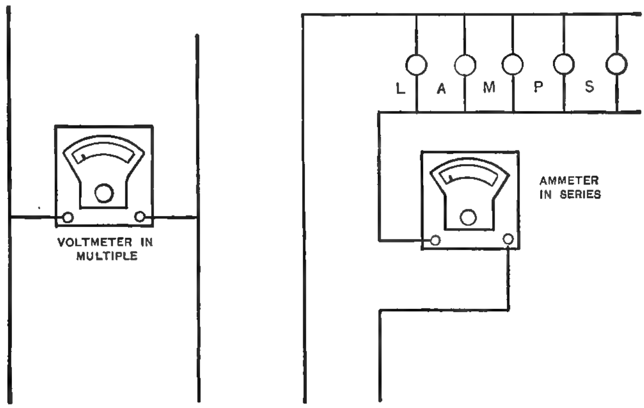


FIG. 26.—Connections of a voltmeter and ammeter.

thus gives a record of the output of power during an hour's, day's, week's, or month's operation. It is an excellent record of the output in power obtained for a given period from a given tonnage of coal.

The circuit-breaker and fuses are of the same class, though of different construction and operation. They both serve to open the circuit (Fig. 27) when an overflow of current occurs. The fuse melts or volatilizes, and thus destroys the continuity of the circuit it connected. The circuit-breaker, through the medium of a controlling electromagnet, opens a switch when an overflow occurs.

Both devices are, in this sense, comparable to the safety-valve of a boiler; this applies particularly in the case of the circuit-breaker, whose operation is electromechanical. As is self-evident, the fuse must be replaced, whenever it blows, with a new and equivalent piece of fusible metal. The circuit-breaker is simply reset by means

of a catch which engages with the armature of the controlling electro-magnet. When that armature moves against the tension or pressure

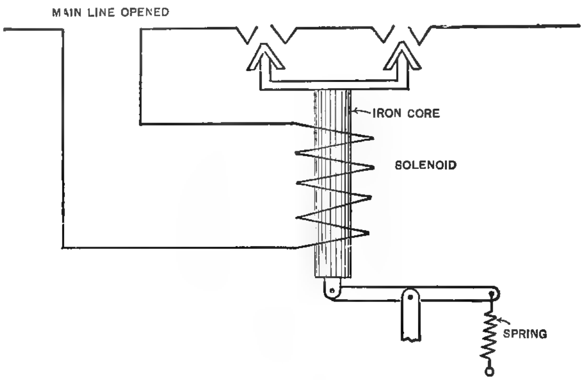


FIG. 27.—A single pole circuit-breaker with action of magnet and spring shown.

of a set-spring—which is impossible unless the current rises above a certain optional value—the catch is released, and the carbon-armored jaws of its switch fly open, breaking the circuit it thus protects.

THE LIGHTNING-ARRESTER

This device is simply an air-gap interposed between the line and the earth (Fig. 28) through the medium of two pieces of metal slightly separated from each other. The idea involved is that static dis-

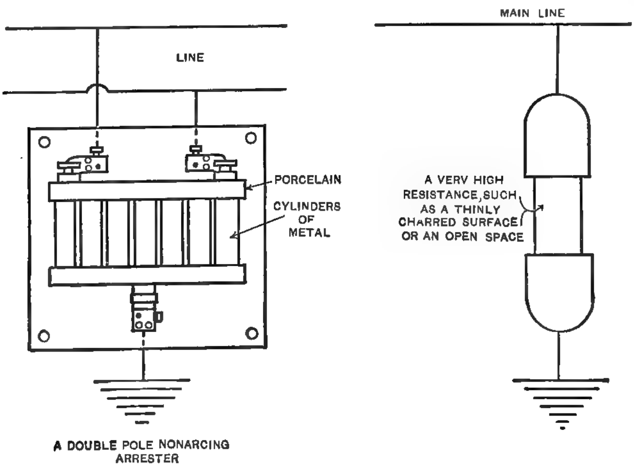


FIG. 28.—Two types of arresters in use.

charges will jump the air-gap to the earth instead of continuing along the conductors. The highly oscillatory nature of the discharge leads to this conclusion, for the reason that a rapidly moving quantity of electricity finds a greater difficulty in permeating a conductor the more rapidly it oscillates.

At a very high rate of oscillation—that of a lightning-discharge in fact—the conductor becomes less conductive than the air. The gap in the arrester thus permits the charge to choose and pass into the earth. A wire leading up from an earth-connection to a piece of metal—the said piece of metal being opposite and near to another connected to a line wire—represents an arrester in its simplest form. Variations of this idea predominate in practice as indicated by many manufactured types.

A GROUND-DETECTOR

As its name implies, this is a device by means of which a ground between the mains, feeders, or branches and the earth is indicated. It is of sufficiently simple construction to warrant no other explanation

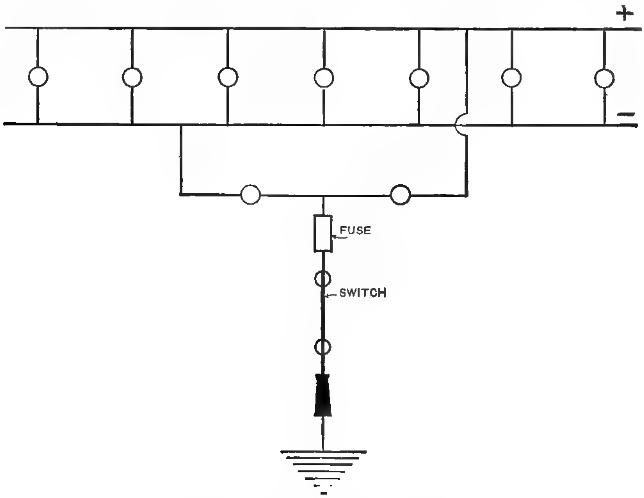


Fig. 29.—Ground-detector.

than that it consists of two lamps in series (Fig. 29) across a 110-volt line. The junction between the lamps is grounded or earthed. When a lamp on one side glows brighter than that on the other, it means a ground on the side of the dim lamp. It is a simple and effective

method of determining the condition of the circuit before the power is turned on.

The existence of a ground, when discovered through the use of this device, calls for immediate investigation and a systematic search. In order to facilitate the test, the feeder-switches should be all opened and the dynamo-mains tested first. If this test does not result in discovering the ground, each feeder-circuit should be thrown in, one after the other, until the lamps again show the discrepancy in illumination noted at the beginning.

The last switch to be closed to effect this result is the one governing the circuit or circuits in which the ground exists. The investigation is then made with respect to the heavier and then with the subsidiary branches of this particular feeder-circuit, until the fixture, chandelier, motor, or other source of trouble is discovered and the fault remedied. A daily test to protect the system is necessary because a ground on both legs, if heavy enough, would constitute a short circuit.

STORAGE-BATTERIES

The storage-cell is employed for a distinct purpose in central-station, power-house, and private-plant work. Its application is best found in central-station and power-house service as a means of averaging up the day- and night-load. If there is a very heavy call made for current—a demand beyond the load-limit of the generators—the storage-battery serves the useful purpose of adding such of its quota as is necessary to meet the demand.

If the demand is frequent but spasmodic there is no substitute for it in an electrical or an economic sense. In other respects the storage-battery, merely as a convenient electrochemical device for transforming electrical energy into chemical energy, is an interesting and commercially useful invention. Its characteristics may be readily comprehended in the following terms:

TYPES OF STORAGE-BATTERIES

There are two types of storage-cell, the Planté and the Faure. The Planté consists of lead plates that have undergone the process (Fig. 30) called "forming," whereby the lead-surface for a considerable

depth has been converted into an oxide of lead. The positive plates—that is to say, the plates always connected with the positive pole of the dynamo—turn into a spongy reddish or chocolate-colored mass. The negative plates, always connected to the negative pole of the charging current, turn gray or slaty in color, due to the development of dioxide of lead.

The peroxide of lead or positive plates and the dioxide of lead or negative plates are thus the recipients of the electricity sent in, storing it up to a certain point, as the popular expression goes, called "the capacity of the plates" or "cell." The solution used is that of a 20-per-cent. sulphuric-acid mixture, or four-fifths water and

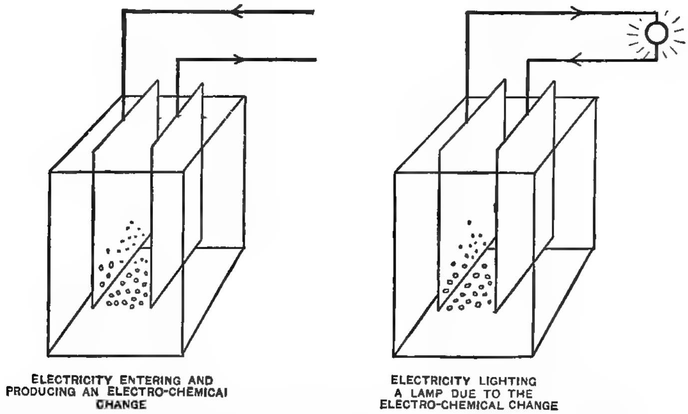


FIG. 30.—Conversion of lead into active material, and then the production of electricity.

one-fifth acid. The acid employed for this purpose is comparatively pure; otherwise distressing and injurious local troubles will develop. The Faure cell differs from the Planté in the respect that the oxide is not formed by a slow electrical process of charging and discharging during a period of many weeks, as was the old and original process, but by mechanically applying the two oxides in the form of a paste. The pasted grid of lead or some lead alloy came into extensive use (Fig. 31), and its interstices were filled with a paste of lead oxide and glycerine. The positive grid originally received a red-lead paste, which, through a comparatively brief forming process, was readily converted into peroxide of lead. The negative plate received a paste

of litharge, a lower oxide of lead, a comparatively brief forming process converting this as well into a dioxide of lead.

The idea in connection with both plates was to get a spongy mass, in close adherence to the grid supporting it, during the process of

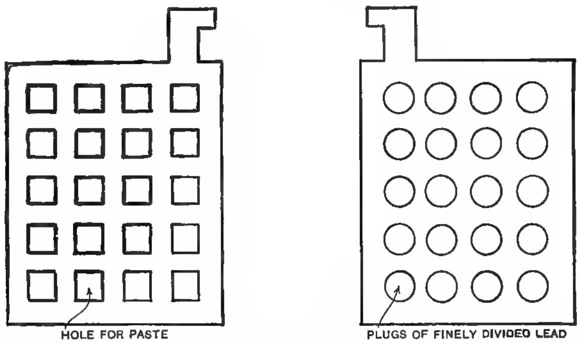


Fig. 31.—Appearance of grid to which paste or a process is applied to rapidly form it for service.

manufacture. The capacity of the plates individually and collectively was thus raised to a certain practical working maximum by which they were rated when sold. This rating is in ampere-hours, a term meaning the number of amperes normal discharge for a certain number of hours. For instance, 100 ampere-hours would mean about 10 amperes discharge for 10 hours; a 250-ampere-hour capacity would mean about 25 or 30 amperes respectively for 10 or 8 hours. The lower the rate of discharge comparatively, the longer the number of hours of service, a sudden heavy call for current beyond the normal rate being likely to cause serious damage to the plates.

The pasted plate and the pure-lead plate are used in conjunction in the following manner: the positive plate is made in the majority of cases of finely divided lead, upon which the electrical action is very rapid. This plate is used in connection with a pasted negative plate. The positive wears out the more quickly, and is therefore replaced the oftener. It must therefore be made as strong as possible because of the peculiar deterioration to which it is subject. The negative plate may outwear the positive two to one or even three to one in some instances.

DIFFICULTIES WITH PLATES

The difficulties with storage-battery plates are at least twofold: First, they are apt to sulphate if left too long in the solution without being well charged. Second, they are likely to bend or buckle under the influence of a very heavy discharge. The sulphating means a hard, flaky, white coating of sulphate of lead (Fig. 32), which is removed only with great difficulty by scraping or by heavy charging and discharging. The acid in this case simply attacks the lead-surface when the plates are well discharged, and starts a distinct chemical action. The moral of this is never to permit a storage-battery to fall very low without recharging.

The battery is generally over 2 volts when normal, and when well emptied of its energy about 1.9 volts or a little lower. This, however, is the limit of discharge per cell for ordinary forms of service. The buckling or bending is caused by the sudden strain on the plates by an inordinate discharge. The rate of discharge which is considered safe is given with each type of cell by the manufacturer, and, if possible, should not be exceeded. The buckling that is so apt to occur in the plates has the effect of loosening the plugs of active material, which may drop between the plates and cause an internal short circuit. It may also make the plates touch at various points unless adequately remedied. The means employed to remedy this difficulty is simply that of making plates sufficiently rigid to withstand the strain without injury. In other words, the plates of modern batteries are made thick enough to remain unaffected throughout their period of usefulness.

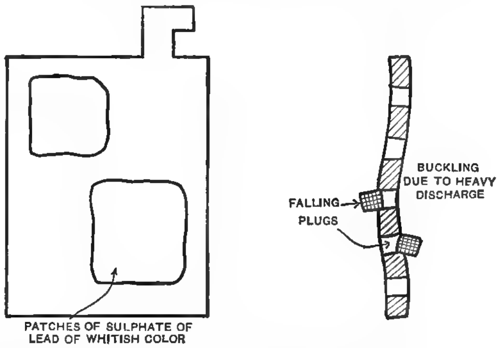


FIG. 32.—Patches of sulphate on surface, and bending of plate.

EFFICIENCY OF STORAGE-CELLS

A comparative test of the efficiency of storage-cells is made by simply sending into each a given amount of power and taking it out again up to a certain point. The voltage of the cells will give a fair idea of the relative value of each cell under the circumstances. But this is not conclusive, as it is necessary to take into consideration the weight of each cell and its period of usefulness in making a fair estimate of its respective qualities.

Cells are made for portable and for stationary use; the weight question is an important one in the first case, though unimportant in the second. Portable cells always deteriorate much quicker than those occupying a fixed position. Each square foot of active plate-surface will give certain maximum and minimum capacities in ampere-hours. Each set of plates will last a certain period of time under the influence of a certain course of treatment. The commercial problem is that of increasing their period of usefulness; the scientific problem is that of increasing their capacity for a given size and weight and of eliminating characteristic defects.

THE BATTERY-ROOM

A battery-room is best designed with reference to ventilation, drainage, heating, water-supply, aisle-space, floor-construction, and absence, as far as possible, of metal-work. The charging of a storage-battery means the development of acid-spray, whose effects are highly deteriorative. Not only must the room be constructed so as to be protected from this evil, but it must be ventilated effectively. Openings near the ceiling at one end, and near the floor at the other end, are the best means of securing a clean atmosphere. The sloping of the floor must be sufficient to thoroughly drain it when wetted through overflow, accumulations, or during the process of flushing it out.

Too much cold is effective in reducing the capacity of the cells; for this reason the battery-room must be kept at a moderate temperature during the winter season. Inspection is necessary at all times, and in order to accomplish this readily the cells must be arranged so as to

be easily accessible. Refilling with water or solution must not be a difficult task in a battery-room. Having the cells low enough down, with an aisle on each side, is a good plan if space permits.

The use of asphaltum paint is a good protection against acid-spray wherever it may deposit; and the floor of the room should be made of vitrified brick laid on concrete and filled in with pitch. To have water conveniently at hand is imperative in a battery-room, though the use of distilled water is far more preferable. If floor-space is limited the cells must be arranged in tiers, each of which must afford enough overspace to readily handle, inspect, and, if necessary, remove defective cells or plates.

CHAPTER XXVII

LIGHTING AND LAMPS

ELECTRIC LAMPS

THE sources of electric illumination have developed sufficiently to represent a distinct department in themselves, and are of equal importance to the system or systems of electric lighting in vogue, with their characteristic accessories. Electric lamps are sufficiently varied in principle and construction to represent a classification of the greatest interest, and they may be divided into the following types:

1. Incandescent lamps that employ a vacuum to protect the incandescent mass in use from the action of the air.
2. Incandescent lamps that do not employ a vacuum, but use an incandescent mass which is inherently inoxidizable.
3. Arc-lamps which employ two carbons that burn in the air and that consume in about eight or ten hours.
4. Arc-lamps which employ two carbons that burn in the air, the carbons having a metal core, which develops a comparatively long arc of unusual light-giving power.
5. Arc-lamps which employ two carbons that burn in a closed globe, through which they last 100 or more hours.
6. Incandescent-vapor lamps that use an incandescent mercury vapor to produce illumination.
7. Incandescent-tube lighting that is effected in long tubes devoid of air, but filled with a highly illuminative gas when affected by a current of the proper pressure and character.

The incandescent lamp with a carbon filament and the closed globe or enclosed arc-lamp are the two most prominent types of lamps (Fig. 33) in use to-day. The arc-lamp which burns with exposed carbons has been modified by the introduction of a metal core and a specially impregnated carbon, and thereby given a new lease of life. It bears the general title of the flaming arc for reasons that will be obvious when it is realized that metallic vapor is effective in this

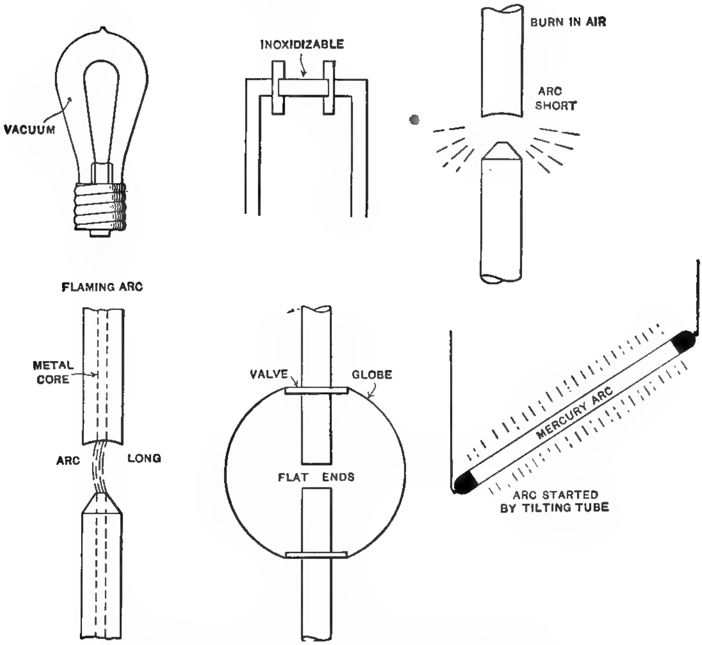


FIG. 33.—Types of electric-light sources in present use.

respect to a marked degree, by enabling an ordinary arc to be elongated sufficiently to give light not only from the carbon terminals, but from the arc itself.

THE INCANDESCENT LAMP

The carbon filament of an incandescent lamp is obtained by carbonizing a fine string of cellulose, enclosing it in a glass globe from which the air has been removed, providing it with platinum leading-in-wires, so that both the metal and the glass will expand and contract together and thus preserve the vacuum, and attaching it to a suitable base for commercial purposes. A lamp of this character is generally used on a 115-volt circuit, and takes a current of from .4 to .5 of an ampere. The value of the lamp commercially is naturally based upon three considerations: (1) The cost in barrel lots; (2) the life in hours of normal candle-power, and (3) the amount of candle-power per unit of power; as, for instance, the number of watts per candle or per lamp of 16 candle-power.

The cost, durability, and efficiency have been the governing influences in developing incandescent-lamp manufacture to its present point of perfection. In the central station or private-plant the efficiency and durability of lamps are points of vital importance. The question in this case is fundamentally that of the cost of the candle-power hours. Incandescent lamps vary in this respect, one producing more candle-power hours at a given power or watt-consumption than another. The lamp lasts from 500 to 600 hours under ordinary conditions, the unit of power adopted being that of 16 candle-power. This light, however, varies in efficiency as the life of the lamp increases. It grows less illuminative with a given amount of power, and therefore becomes less efficient. In fact, the net conclusion inevitably reached is that old lamps are very wasteful (Fig. 34), as may be readily shown as regards the light they give and the watts they consume. For instance, a new 16 candle-power lamp will use .4 of an ampere and 115 volts, or $.4 \times 115 = 46.0$ watts. At the end of 600 hours it will require 55 or 60 watts to give the same light.

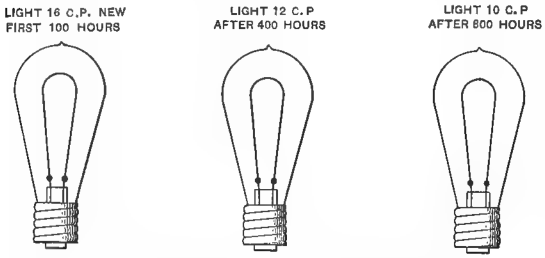


FIG. 34.—Relative light of old and new lamps.

The fact that the old lamps do not break is a temptation to use them; but the difference between 60 and 46 watts is 14, or nearly $33\frac{1}{3}$ per cent. more power. If the pressure is not raised the old lamps will give no more than 10 or 12 candle-power, thus causing a waste either way of practically $33\frac{1}{3}$ per cent. of the fuel. Old lamps or inefficient lamps are simply coal-wasters, which cost more in fuel than it would cost to buy new lamps.

As the primary purpose of a plant is to supply a certain amount of light, it seems self-evident that the keeping of it up to a normal value is a responsibility that cannot be carried out without adequate means. For this reason the efficient lamp is a saving, because it means not only less coal, but less wear and tear of machinery, less rate of depreciation, in fact, in providing satisfactory lighting. With respect to the arc-lamp, 10 amperes and 115 volts in incandescents

will light about 20 or 25 lamps of 16 candle-power, or produce about 300 or 400 candle-power. An arc-lamp of the enclosed type, taking the same watts, will produce from 1,200 to 2,400 candle-power, the ratio of light produced being as 4 or 6 is to 1 in favor of the arc as far as efficiency is concerned.

THE NERNST LAMP

This lamp, in which a piece of rare oxide burns in the open air, is more efficient than the incandescent lamp so called, because the temperature of the incandescent mass is raised so much higher. In other words, the whole question of efficiency in incandescent lamps hinges upon that of temperature. If a carbon filament could stand the temperature at which the rare oxide burns (Fig. 35), the efficiency would double. Instead of taking from 3 to 4 watts per candle-power, it would take from 1.5 to 2 watts. But carbon will not stand this heat under ordinary circumstances for any length of time.

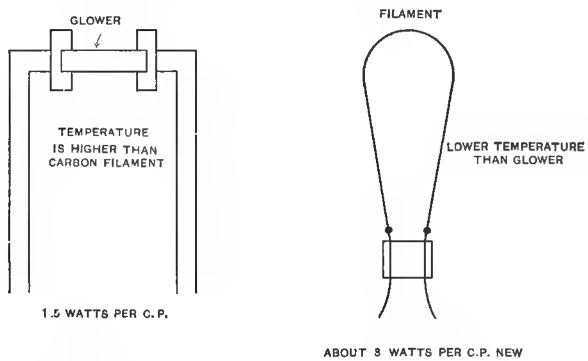


FIG. 35.—Power consumed for light at higher temperatures.

The lamp mentioned above, however, using the rare oxide in air, requires that this oxide be primarily heated before sufficient current will pass to heat it individually. An automatic heater is therefore used for this purpose. The filaments of these lamps are technically called "glowers," of which one, two, or more may be used in a given case. The light is produced at the rate of about 1.5 watts per candle-power, or at twice the average efficiency of new incandescent lamps.

THE OPEN ARC

By the term "open arc" is meant the type of lamp in which the carbons burn in the open air. In this type the carbon tips and the arc combine to produce light. The tips are in some instances the most effective, particularly in the enclosed type. The arc gives an average rated spherical candle-power of about 2,000 with a current of 10 or 12 amperes and about 50 volts. This means two lamps in series on a 115-volt circuit. For high-tension lighting the lamps are arranged in series, 2,000 volts being sufficient to light forty lamps in series. The difficulty and expense are found in the removal of the carbons, their cost, and the general attention required.

THE FLAMING-ARC LAMP

The accentuation of the light and length of the arc itself, as previously stated, is a source of light which, bearing the descriptive name of the "flaming arc," has proved exceedingly efficient as an outdoor

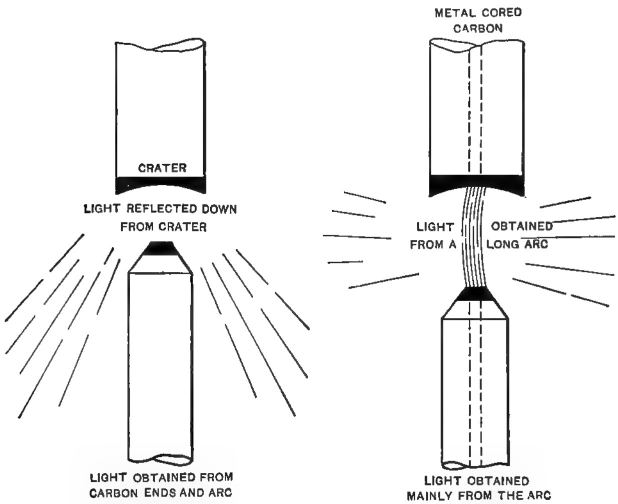


Fig. 36.—Principle of the ordinary and the flaming arc lamp.

illuminant. The production of gases limits its use for indoor illumination, except in such cases where the ventilation is excellent, thereby rendering the gases undetectable. The lamps burn two in series, on a

115-volt circuit, the carbon tips (Fig. 36) as well as the flame of the arc, with its metallic constituents, developing an enormous illumination —at least three or four times greater than that of the ordinary arc as far as effective light is concerned.

The carbons are replaced about every day, and the lamp inspected and readjusted no oftener than the ordinary open-arc lamp. A resistance in series is employed, placed in the upper part of the lamp, to limit the current when supplied with a higher voltage than necessary. The carbons in this particular type are held at an angle with the vertical plane, thus equably reflecting the light from the flaming arc as well as permitting it to be well distributed spherically. By means of the carbons the light developed is of a golden or a reddish tinge. The candle-power is about 4,000 per lamp or over, and is remarkably effective on account of its peculiar quality due to the salts used to impregnate the carbons.

THE ENCLOSED ARC

This is an open arc with the carbon-ends adjusted to a globe supplied with an outlet-valve. The oxygen is quickly burnt up, and the burnt air (Fig. 37) in consequence has little or no effect upon the carbons, which thus last 100 or 150 hours instead of 10 or 15.

A saving in carbon and care is thus in evidence, which is counter-balanced, however, by the deposits on the globe and the cutting down of the light. These lamps take about 100 volts, instead of 50, on account of the length of the arc, and are therefore heavy users of current and pressure on 115-volt circuits. The economic question is that of evaluating the cost of carbons and labor against the cost of extra power. Into this estimate the consideration of safety must enter on account of the closed character of the lamp and the resultant candle-power.

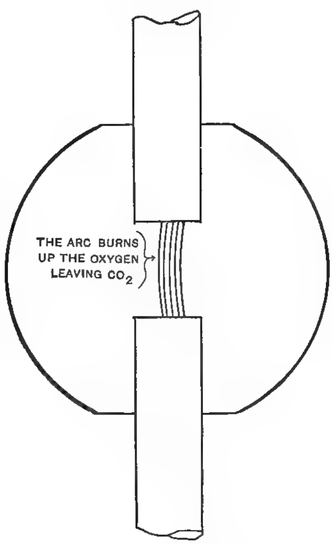


Fig. 37.—The carbons burn with flat ends.

MERCURY-VAPOR LAMP

In this lamp a tube with electrodes forms an arc of mercury, in which band of dazzling light, however, all red rays are missing. This defect exhibits itself in the development of ghastly flesh-effects, green and blue (Fig. 38) causing a livid appearance of the lips, face, and hands. For this reason this lamp cannot be used for domestic or ornamental lighting, but is serviceable for docks, factories, ware-

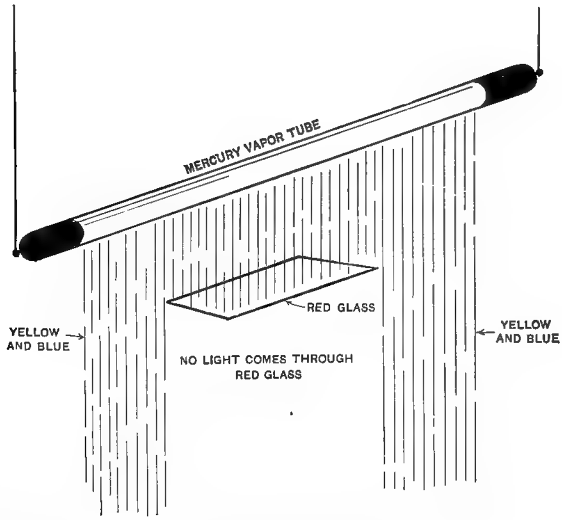


Fig. 38.—Experiment with red glass—which can only transmit red rays—to prove the absence of red light.

houses, etc. The tube is automatically tilted to start the arc by permitting a thin stream of mercury to volatilize between the electrodes. The efficiency of this lamp is high enough to establish its commercial value on a permanent basis.

VACUUM-TUBE LIGHTING

The use of a 150-foot tube, through which an alternating current passes, is a development of Geissler's early experiments on a commercial scale. The so-called vacuum, in conjunction with certain gases, is a source of illumination which has found a place in daily practice. The absence of wires in all but one spot where the tube enters and leaves

after making its circuit of the room, is an interesting feature. The length of tube is suspended (Fig. 39) by brass clasps, and the light is estimated from the candle-power per inch.

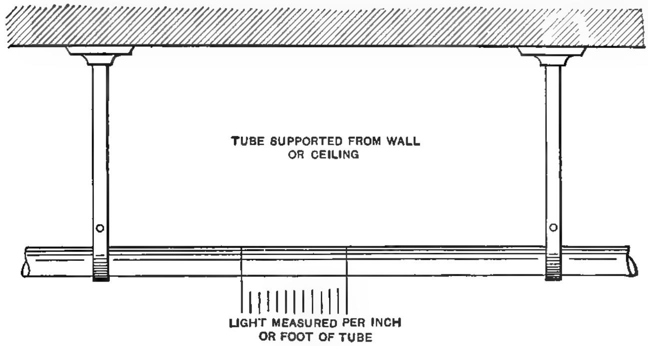


Fig. 39.—Vacuum tube lighting in daily practice.

The power used is alternating, and is transformed into a current with a particular form of wave. The action of this upon the gas in the tube causes a uniform glow of the pleasantest character. No brilliant centre of light appears, but a type of diffused radiance that is highly suitable to indoor illumination. The efficiency of this system is claimed to be greater than that of the incandescent lamp.

ELECTRIC-LIGHT EQUIPMENTS

There are many methods in use of obtaining systematic rotation in a mechanical sense; and it is quite evident that rotation of this character is suited in every respect to electric lighting through the medium of the dynamo. The well-known devices producing this type of power are steam-engines or turbines, gas- or oil-engines, and water-wheels. Each deserves separate consideration in a treatment, however brief, of this subject.

STEAM ELECTRIC PLANTS

The consumption of steam in reciprocating engines or turbines obtained from boilers represents in total the elements of a steam-plant. The boiler and its accessories, the engine or turbine, and

the generators and switchboard, with its light- or power-circuits, comprise the modern equipment whose size and character of service give it the name of private plant, central station, or power-house. The fact that the electricity is consumed privately, does not necessarily limit the size of the plant. A private plant may be far greater than a central station, yet differ from it in purpose and hours of service. A central station is a public dispenser of light and power, operating under a municipal franchise. A power-house, in contradistinction, is a street-railway equipment, sending out power primarily for the cars along the route, yet incidentally supplying electricity for light and power.

These three great equipments are thus defined as the private plant, however large, having no franchise; the central station, generating electricity for light and power purposes; and the power-house or street-railway plant, whose franchise is directly intended to have it serve street-railway interests. The general efficiency of these equipments is dependent upon the character of the boilers, engines or turbines, and generators in operation. The weight of coal consumed per horse-

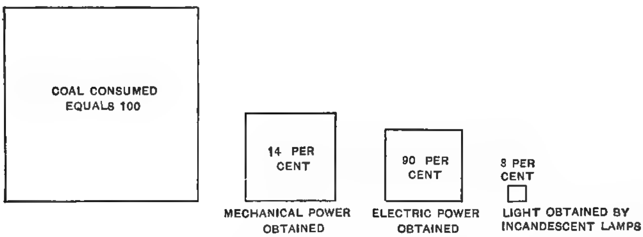


Fig. 40.—Relative light effect obtained from a given amount of coal by electric lighting by incandescent lamps.

power or kilowatt hour and the cost of handling that power until it is paid for by the consumer constitute the economic problem presented to the manager of large or small equipments of the central-station class.

The range of efficiency for the steam, generating, and transmitting and distributing sections are all well known, as likewise that for the various types of lamps. These facts may be arranged (Fig. 40) in a convenient form for reference.

1. The steam section has an efficiency of from 14 to 16 per cent. from the coal to the mechanical energy delivered.

2. The generating section has an efficiency of from 90 to 95 per cent. from the engine or turbine to the switchboard.

3. The transmitting and distributing section has an efficiency of from 90 to 95 per cent. from the switchboard to the consumers' lights.

4. The illuminating section, or lamps, have an efficiency of from 3 to 10 per cent. from the circuit terminals to the candle-power produced. The rating would be about 3 per cent. for incandescents, 6 per cent. for Nernst and mercury vapor, and about 10 per cent. for arc-lamps in general.

It seems a very difficult matter at present to generate light-waves without first developing heat-waves, as, for instance, in all the illuminants known, with the possible exception of the vacuum-tube. The making of light is therefore restricted by the limit of present scientific knowledge. The only gain, outside of the element of depreciation—which is reducible only by strengthening or by reducing the number of deteriorating parts in a plant—is by cheapening the original power-supply. This is a condition implied by the fixity of the general efficiencies in all but the starting-point. Here the water-power proposition becomes of interest, as well as that concerning the development of power from explosive engines.

WATER-POWER PLANTS

A stream of quickly flowing or falling water, developing enough energy to move a water-wheel or turbine all the year round, is an interesting possibility if it is near a city or town. If many miles away from a large community, its usefulness will be governed by its power. It pays to transmit enough power thus obtained, if the supply is comparatively regular. Otherwise its value is limited, as when the supply varies greatly from summer to winter or ceases altogether temporarily.

A variable source of power may mean an auxiliary steam-plant, making the economic issue doubtful in the extreme. Power thus obtained, however, is cheap if regular, and simplifies the electric-light-and-power proposition, provided the distance of transmission is not so great that the investment for poles, insulators, and conductors represents an unreasonable figure.

A turbine or water-wheel varies in efficiency from 70 to 85 per cent. The delivery of that power at a distance costs, roughly, in proportion

to the distance. This is a problem best solved by reference to existing data concerning similar power-transmission plants. Hydro-electric plants, as they are called, really consist of only the water-wheel, the generator, the switchboard, and the outside circuits. Cheap power is the natural consequence of an equipment of this character if intelligently constructed and handled.

GAS-ENGINE ELECTRIC PLANTS

Instead of burning coal the process of distilling it for its explosive gases, and using them as a source of power, is becoming prevalent. Small gas-making plants of this character are called "gas-producer plants." The government tests show an immense increase in fuel-efficiency in distilling coal and instead of burning the coal under a boiler, exploding the gas to gain power in a gas-engine. The use of so-called illuminating-gas in a gas-engine, and the application of the resulting power to the production of electricity, is becoming more emphasized in central-station practice than ever before. Private plants thus equipped are numerous on account of the elimination of the boiler and its accessories and the consequent simplification resulting. If, instead of the boiler, a gas-producer plant is installed wherein the power-supply is to be great enough, the expense for gas is so reduced that a kilowatt hour costs less than one-half of its production in a steam-plant of equal size.

The depreciation of gas-engines and their accessories is therefore balanced up against steam-engines and their accessories in forming a correct estimate of the cost of operation. The capacity, horse-power, speed, and weight of a line of gas-engine plants for small installations are given in the following table, with the form of the manufacturers' guarantee:

Kilo-watts.	Horse-power.	Speed.	Type.	Weight of engine.	Floor-space, inches.	Weight of direct-connecting unit.	Floor-space, inches.	No. lights, 16 candle-power.
$2\frac{1}{2}$	6	400	Vertical.	2,050	36×39	2,850	36×64	50
7	12	360	2 cylinder.	3,200	38×42	5,700	38×96	120
10	18	350	2 cylinder.	4,400	40×44	7,200	40×102	180
20	30	300	2 cylinder.	8,200	53×57	13,500	53×122	360

MANUFACTURERS' GUARANTEE.

> We install these plants with the guarantee that there will be no noise from the exhaust. We guarantee every machine against breakage or undue wear for one year.

To recapitulate, with reference to the foregoing facts the present practice shows the limit of power- and light-efficiency, and indicates, as a means of improving the net efficiency, the necessity for either cheapening the power or changing the lighting system in vogue, not superficially, but fundamentally. Cheapening the power is an obvious way, relatively, but this is not true of the light. It must be understood that because of the comparatively low efficiency of the light, even of the flaming arc, it becomes imperative to make power cheaper to cheapen electric lighting.

The intermediate machinery between the heat or gas-explosion and the light will probably remain unchanged for some time. Progress therefore will be best evidenced scientifically, and subsequently commercially, by the development of a plan or system by means of which the long heat-waves are cut entirely out, and the short light-waves are produced with greater directness. This would mean cold instead of hot light at the source, and an immense saving in energy now uselessly and widely dissipated.

QUESTIONS AND ANSWERS ON CHAPTER XXIV

Question.—How may the operation of the dynamo be best described?

Answer.—As the movement of conductors through lines of force, or the movement of lines of force through conductors.

Question.—How do the electromotive forces of a motor and dynamo serve different purposes?

Answer.—The electromotive force generated in the armature of a motor is opposite to the electromotive force sending the current in, and acts as a regulator. The electromotive force of a dynamo is used to send the current through the circuits and their resistances.

Question.—What is produced in conductors cutting lines of force, or in lines of force cutting conductors?

Answer.—Electromotive force is produced within the conductors.

Question.—How does the alternator and the direct-current generator differ?

Answer.—The direct-current generator uses a commutator in order to send out a current flowing always in the same direction. The alternating-current generator uses collector-rings, which permit all the alternations generated within the armature to occur outside in connected circuits.

Question.—What is the formula for calculating electromotive force?

Answer.—The volts generated equal lines of force $\times$ revolutions of the armature per second $\times$ the armature-conductors $\div$ 100,000,000.

Question.—What reverses the direction of a current in a conductor?

Answer.—The fact that it is being moved past a north pole or a south pole. The electromotive force tends to send a current in one direction when the conductors pass a north pole, and in the reverse direction when they pass a south pole.

Question.—What is the action of the commutator and brushes?

Answer.—To permit all positive impulses to flow into one brush or set of brushes, and all negative impulses to flow into the other brush or other set of brushes.

Question.—Where do the positive and negative impulses of current come from?

Answer.—From conductors which pass the north and south poles respectively in a two-pole or multipolar field, and with which commutator bars are connected; these bars transmit the positive and negative currents to the brushes.

Question.—What kind of current is naturally generated in a two-pole or multipolar direct-current generator?

Answer.—A series of reversing electromotive forces, or what is called an alternating current, which is rectified or commutated.

Question.—How are dynamos classified with respect to the character of their currents?

Answer.—As alternating- and direct-current generators. The direct-current machines are further classified as series-. shunt-, and compound-wound generators.

Question.—Of what use is the transformer?

Answer.—To raise or lower the voltage; the volts are raised or stepped up for transmission, and lowered or stepped down when the current is to be distributed.

Question.—How is the field regulated, and what is the effect of this regulation on the voltage?

Answer.—The field is regulated by means of a resistance in series. When this resistance is increased or decreased the current in the fields decreases or increases. A more powerful field means more lines of force and more volts, and a weaker field means less lines of force and less volts.

Question.—Upon what principle does the series-wound dynamo regulate to preserve a constant current and a varying potential?

Answer.—Upon the principle that some armature-conductors produce more volts than others when in certain positions in the field. In consequence the brushes may be made to touch either where the pressure is high or low by an automatically controlling electromagnet in series with the line.

Question.—How is a shunt dynamo regulated for a constant potential and a varying current?

Answer.—By controlling the current entering the fields the shunt dynamo is made stronger or weaker, thus enabling the armature to produce a higher or a lower voltage to compensate for the armature-reaction in the form of drop and a reactive field.

Question.—How is regulation accomplished in a compound-wound dynamo to preserve a constant potential with a varying current?

Answer.—By means of an adjunct coil in series with the main line, the increasing current of the armature enables the dynamo to produce more magnetism. This magnetism is so regulated that its increase approximately counterbalances the loss in magnetism sustained by the reactive effect of the armature. It also supplies enough extra lines of force to enable the armature to generate as many more volts as are needed to make up for the drop within its conductors.

QUESTIONS AND ANSWERS ON CHAPTER XXV

Question.—What two cases are presented with respect to grounds?

Answer.—A ground in one or more coils.

Question.—What is the effect of a heavy ground?

Answer.—Heat in the coil or coils thus affected.

Question.—How is a short circuit explained?

Answer.—As a case in which the current enters a path of lower resistance.

Question.—What is done to dispel moisture?

Answer.—The coil, if possible, is carefully baked.

Question.—Name some of the causes of sparking.

Answer.—Among the principal causes of sparking may be mentioned: too great a load, a wrong position of the brushes, loose brushes or projecting mica, a general design that is poor (such as a badly proportioned air-gap), etc.

Question.—When should a hard or a soft brush be employed?

Answer.—When the commutator is hard-drawn copper the carbon-brush should be used. When the commutator is of softer metal a brush must be used that will not grind the commutator into metallic powder.

Question.—When is sparking ineradicable?

Answer.—When the design is bad, by which is meant that the relationship between the thickness of the air-gap, the arc of the pole-piece, and the load the armature bears, is not correct.

Question.—At what instant does sparking generally occur?

Answer.—When the conductor enters the lines of force of a new pole after leaving one of opposite polarity.

Question.—What is the purpose of the pole-piece fringe?

Answer.—To permit a sparkless reversal of the current.

Question.—If the dynamo will not generate, what are the causes which may be regarded as effective?

Answer.—There may be no residual field; the magnets may have the same polarity; the field-coil may have an open circuit; the armature-connections may be open or may be short-circuited, etc.

Question.—What causes black bars in the commutator?

Answer.—A broken coil in the armature causes black bars between the ends of the break.

Question.—What are the causes of a hot armature?

Answer.—An overload, short-circuited coils on the armature, and very thick conductors.

Question.—What are the types of direct-current motors that are in use?

Answer.—The series, the shunt, and the compound wound.

Question.—What causes heat in the commutator?

Answer.—Bad contact between the brushes and the commutator, a small commutator, too much pressure from the brushes, or a brush of too great a resistance.

Question.—What is the effect of increasing or diminishing the current?

Answer.—The heat varies as the square of the current in amperes. Twice the amperes means four times the heat, three times the amperes nine times the heat, etc.

Question.—What general rule may be depended upon with regard to heated bodies, whether heated by electricity or by any other means?

Answer.—That the amount of external or radiating surface will govern the rise of temperature with any steady supply of heat.

Question.—In what important respect do steam and electrical practice differ?

Answer.—In steam practice the effort is made to prevent the heat from radiating; in electrical practice the radiation of the heat from conductors is obligatory.

Question.—What will happen to a series motor with no load?

Answer.—It will wreck itself by attaining a destructive speed.

Question.—How is a shunt-wound motor started up?

Answer.—Always with the field full on, and a rheostat in series with the armature.

Question.—What is the purpose of a compound winding in a motor?

Answer.—To either strengthen the field, or weaken it with a heavy load.

Question.—With what effect will a stronger or a weaker field operate, and what are their uses?

Answer.—A stronger field makes a shunt motor run slower, and a weaker field faster, with a given load. A compound winding permits this through the influence of the series coil, which may be connected so as to strengthen or weaken the field.

Question.—What is the value of the back electromotive force of a motor?

Answer.—It is equal to the difference between the impressed electromotive force and the voltage required to send the amperes at any particular point of load through the armature.

Question.—What is the difference in a direct-current motor between doing the greatest possible amount of work and having a high efficiency?

Answer.—When making the motor do its greatest possible work it will be overloaded, run slowly, and thus develop power at a very low efficiency.

Question.—What is an indication of an open circuit in the armature?

Answer.—A high speed, a line of sparks or two burnt bars.

Question.—What is the back electromotive force of a motor?

Answer.—The electromotive force developed by the armature-conductors cutting the lines of force.

Question.—What is the electrical efficiency of a motor?

Answer.—The ratio between the back and the impressed electromotive force.

Question.—What is the commercial efficiency of a motor?

Answer.—The ratio between the mechanical power taken out and the electrical power sent in.

Question.—What are the causes of noise in motors?

Answer.—The armature-slots and the pole-tips, the brush on the commutator not being properly inclined, a rough commutator, and lack of lubrication.

QUESTIONS AND ANSWERS ON CHAPTER XXVI

Question.—What is meant by the term "switchboard"?

Answer.—Technically a board on which switches, protective apparatus, measuring-instruments, etc., are found, and by means of which the entire system is operated.

Question.—What are the classes of devices found on a switchboard?

Answer.—Measuring-instruments, controlling devices, protective devices, regulating devices, and testing devices.

Question.—How are the circuits classified?

Answer.—As mains, feeders, branches, and sub-branches.

Question.—What is a centre of distribution?

Answer.—A point on each floor, or group of floors, from which control over those particular circuits may be exercised.

Question.—What is meant by the term "panel board"?

Answer.—Small boards set in the wall that are supplied with switches and fuses for a given group of circuits.

Question.—What fundamental form of protection must be supplied to circuits in all cases?

Answer.—Protection against overflows of current in the form of fuses or circuit-breakers.

Question.—What are the natural parts or sections into which a switchboard may be subdivided?

Answer.—Generating, feeding, and metering sections.

Question.—What do the ammeter and voltmeter indicate?

Answer.—The ammeter indicates the extent of the load on the dynamo. The voltmeter indicates the pressure at which the current is supplied.

Question.—How do these two instruments differ in design?

Answer.—The ammeter is made of as low a resistance as possible, the voltmeter of as high a resistance as possible.

Question.—How are ammeters and voltmeters connected up in circuit?

Answer.—The ammeter is always placed in series with the load, the voltmeter always in multiple.

Question.—Of what purpose is a wattmeter in a circuit?

Answer.—It indicates the total of power used in any period of time, as given in watts or kilowatt hours.

Question.—How does the circuit-breaker operate?

Answer.—It opens the circuit abruptly through the medium of a switch and controlling electromagnet, when the current reaches a given value to which the magnet is set to operate.

Question.—What governs the action of a lightning-arrester?

Answer.—The oscillatory nature of the discharge, through which it finds less resistance in an air-gap than a conductor under certain conditions.

Question.—What is a ground-detector?

Answer.—A device by means of which the contact between a leg of the circuit, or both legs, and the earth, directly or indirectly, is indicated.

Question.—What would be the result of a heavy double ground?

Answer.—A heavy double ground is the equivalent of a short circuit. If not heavy, it constitutes a source of leakage under normal conditions.

Question.—What is the storage-cell used for?

Answer.—For the purpose of creating a higher average load-line in lighting and power service. It also serves to temporarily increase the capacity of the plant.

Question.—What are the two types of storage-cell that are employed?

Answer.—The lead plate or Planté and the pasted grid or Faure type.

Question.—With which plates are the positive and with which the negative pole of the generator connected?

Answer.—The positive pole is always connected with the reddish plates, and the negative pole with the gray plates.

Question.—What is the nature of the solution employed?

Answer.—A 20-per-cent. sulphuric-acid solution.

Question.—What is the meaning of capacity in a storage-battery?

Answer.—The capacity as measured in ampere-hours, or the number of amperes of current for a certain number of hours.

Question.—Which is better, a low or a high rate of discharge from a storage-battery?

Answer.—A low rate is better than a high, as the durability of a stationary plant is thus increased.

Question.—What are the main difficulties with storage-cells?

Answer.—Weight (if used for locomotion), sulphating, and buckling.

Question.—What is the cause of sulphating and of buckling?

Answer.—Sulphating is due to the discharged plates being left in the acid solution too long uncharged. Buckling is due to the warping or twisting of the plates if too thin or if the discharge is too heavy.

Question.—What is the efficiency of a storage-cell?

Answer.—The percentage obtained by dividing the watts taken out by the watts sent in.

Question.—Which would govern the choice of a certain type of cell, high efficiency and weakness or lower efficiency and greater durability?

Answer.—The durability of the cell means a greater advantage than its higher efficiency without it.

Question.—What governs the design of a battery-room?

Answer.—Ventilation, drainage, heating, water-supply, aisle-space, floor-construction, and no metal-work.

Question.—How does cold affect the cell-capacity?

Answer.—It lowers its capacity, and for that reason the battery-room must be kept reasonably warm.

Question.—What paint is durable in a battery-room?

Answer.—Asphaltum paint, or a paint not affected by acid or the fumes of charging.

Question.—How should the cells be arranged?

Answer.—They are best arranged in tiers one above the other, with plenty of space for lifting out or examining the cells.

QUESTIONS AND ANSWERS ON CHAPTER XXVII

Question.—How are electric lamps classified?

Answer.—As incandescent, arc, mercury vapor, and vacuum tube, which, with their variations, include the total field.

Question.—What are the two most prominent types of lamps in use?

Answer.—The incandescent with carbon filament, and the arc of the enclosed and the open type.

Question.—What are the features of importance in the so-called flaming-arc lamp?

Answer.—The use of an impregnated and cored carbon and the greater extension of the arc so as to give greater illumination.

Question.—How is the modern lamp-filament made?

Answer.—By the carbonization of a fine thread of cellulose, to which are attached platinum leading-in wires sealed in a glass stem.

Question.—How are lamps rated?

Answer.—By the watts or power consumed per candle-power.

Question.—Why is platinum employed as leading-in wires?

Answer.—Because the platinum expands and contracts at the same rate as the glass, and thus preserves the vacuum.

Question.—What are the three elements to consider in lamps?

Answer.—The cost, durability, and efficiency.

Question.—What is the difference between old lamps in service and new?

Answer.—New lamps take less watts per candle-power than old ones. The older a lamp the more watts consumed per candle-power; hence old lamps are less efficient and more wasteful commercially than new.

Question.—How does the light of an arc-lamp compare with that of an incandescent lamp from the standpoint of efficiency and candle-power?

Answer.—An incandescent lamp takes about 3 to 3.5 watts per candle-power; therefore 1,000 candle-power obtained in this manner means a consumption of from 3,000 to 3,500 watts. An arc-lamp takes about 600 watts to deliver about 1,200 candle-power, or $\frac{1}{2}$ watt, or even less, per candle-power, depending upon the type of lamp. On this basis 1,000 candle-power by incandescent means at least 3,000 watts consumed; by arc it means about 500 watts consumed.

Question.—What is a Nernst lamp?

Answer.—A lamp in which a piece of rare oxide is raised to incandescence by means of a current.

Question.—Wherein does a Nernst lamp possess advantages over the carbon filament?

Answer.—In the fact that it can be safely raised to a higher temperature; also in the fact that no vacuum is necessary.

Question.—What is an open arc?

Answer.—An arc-lamp whose carbons burn in the open air.

Question.—How are open-arc lamps connected up?

Answer.—Generally as two in series on a 110- or 115-volt line.

Question.—How are they connected on high-tension arc-lines?

Answer.—In series throughout the system. An allowance of about 50 volts per lamp is made, which would mean at least forty open arcs in series on a 2,000-volt system.

Question.—What are the features of the flaming-arc lamp?

Answer.—It produces a more efficient light than other arcs; it throws off deleterious gases; its carbons must be renewed every ten or fifteen hours; its carbons are set at an angle to each other; the flame gives out light as well as the tips.

Question.—What are the features of the enclosed-arc lamp?

Answer.—It will burn 100 to 150 hours without attention or carbon-renewals; it takes more power than an open arc; its carbons last because the oxygen in the small globe surrounding the tips is consumed, and a valve prevents more air from entering freely.

Question.—What defects appear in the enclosed arc?

Answer.—The deposits on the inner globe cut down the light, and the separation of the carbons calls for a higher pressure than the open arc, a pressure of from 80 to 100 volts being required. This means only one lamp on a 110- or 115-volt line, and power wasted in the resistance necessary to limit the current.

Question.—What is a mercury-vapor lamp?

Answer.—A lamp formed by a tube with terminal electrodes, and containing a small quantity of mercury. By tilting the tube the mercury connects the electrodes, vaporizes, and forms a band of dazzling light.

Question.—What are its features in practice?

Answer.—The production of a form of light without any red rays, a high efficiency, an automatic device for tilting it to start the arc.

Question.—What is a vacuum-tube system of lighting?

Answer.—At present it means a long tube of 50, 100, or 150 feet, containing a gas which, when affected by a high-pressure alternating current, produces commercial lighting effects. It originally represented a tube containing a vacuum, with electrodes by which the residuum of air caused illumination by molecular bombardment.

Question.—What are its features?

Answer.—An efficiency claimed to be that of the incandescent lamp, no deterioration of any consequence, ready installation, no glare, but a uniform glow.

Question.—What are the sources of power employed for electric lighting?

Answer.—Steam, gas, oil, water, and in some instances the sun or the wind.

Question.—What are the elements of an electric-light plant?

Answer.—The boiler and its accessories, the engine or turbine, the switchboard, and the circuits inside and outside used for distribution and transmission.

Question.—What is a private plant, a central station, and a power-house?

Answer.—A private plant supplies current to a building or buildings and grounds, without any sale of current taking place or any municipal relationship being involved. A central station has as its primary purpose the sale of current under a franchise granted by the community to be supplied. A power-house is generally representative of a plant supplying street-railway power to a trolley-line. It may be the distributing centre of a transmission-plant miles away.

Question.—Into what sections may a plant of public or private utility be divided?

Answer.—Into the steam section or power-developing element, the generating section, the transmitting and distributing section, and the illuminating and motive-power section.

Question.—What is the great difficulty with our present method of light-production?

Answer.—The method of generating heat-waves in order to reach the light-waves required.

Question.—What are the advantages, in an economic sense, of a water-power source instead of steam?

Answer.—The limited cost of the power, provided it can be depended upon throughout the year, and the high efficiency of the water-wheel.

Question.—What is the economic disadvantage of a water-power plant?

Answer.—The limited value of the water-power due to its great distance from the community, and the cost of transmission, even though the power is regular, eliminating the gain otherwise in evidence.

Question.—What are the advantages of producer gas in electric-light plants?

Answer.—The use of the fuel in such a manner that the electricity produced is cheaper than by means of direct combustion for steam. No boiler is required, but a gas-producing plant, which distills the

coal or other fuel, and the gas thus obtained is exploded in a gas-engine. Instead of heat and steam-power, use is made of gas and the explosive force latent when mixed with air.

Question.—What is the drift of engineering practice in the lighting field?

Answer.—The production of cheaper power, in order to not only cheapen the cost of light, but increase the quantity in use per capita.

Question.—What is the tendency in scientific light-making?

Answer.—To attempt the production of light that possesses no preliminary heat-waves, in order to gain an efficiency that will increase the amount of light obtainable from a given power-consumption many fold. Where an incandescent lamp wastes 97 per cent. in heat, and gives out only 3 per cent. in light, the elimination of the heat-element would mean theoretically 33×16, or 528 candle-power instead of only 16 candle-power for a watt consumption of 50.

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